TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 40

FORTIETH ANNUAL MEETING
NEW YORK, N. Y., FEBRUARY 5-9, 1934

FORTIETH SEMI-ANNUAL MEETING
BUCK HILL FALLS, PA., JUNE 19-22, 1934

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
51 MADISON AVENUE
NEW YORK, N. Y., U. S. A.

PRINTED IN U.S.A.

CONTENTS

CHAPTER PAGE

Officers and Council—1934

American Society *of* Heating *and* Ventilating Engineers

President .... C. V. Haynes
First Vice-President .... John Howatt
Second Vice-President .... G. L. Larson
Treasurer .... D. S. Boyden
Secretary .... A. V. Hutchinson

Council

C. V. Haynes, *Chairman* John Howatt, *Vice-Chairman*

One Year	*Two Years*	*Three Years*
Albert Buenger	R. H. Carpenter	M. C. Beman
J. F. McIntire	J. D. Cassell	E. H. Gurney
F. E. Giesecke	L. W. Moon	W. A. Russell
W. T. Jones	F. C. McIntosh	O. W. Ott
W. E. Stark		

Committees of the Council

Executive: John Howatt, *Chairman;* J. D. Cassell, E. H. Gurney.

Finance: R. H. Carpenter, *Chairman;* F. C. McIntosh, J. F. McIntire.

Meetings: W. E. Stark, *Chairman;* F. E. Giesecke, L. W. Moon.

Membership: F. C. McIntosh, *Chairman;* Albert Buenger, W. A. Russell.

Advisory Council

W. T. Jones, *Chairman;* W. H. Carrier, Homer Addams, R. P. Bolton, S. E. Dibble, W. H. Driscoll, H. P. Gant, John F. Hale, L. A. Harding, H. M. Hart, E. Vernon Hill, J. D. Hoffman, S. A. Jellett, D. D. Kimball, S. R. Lewis, Thornton Lewis, J. I. Lyle, J. R. McColl, D. M. Quay, C. L. Riley, F. B. Rowley, F. R. Still *and* A. C. Willard.

Committees—1934

COMMITTEE ON RESEARCH

John Howatt, *Chairman*
F. E. Giesecke, *Vice-Chairman*
Dr. A. C. Willard, *Technical Adviser*
F. C. Houghten, *Director* O. P. Hood, *Ex-Officio Member*

One Year	*Two Years*	*Three Years*
D. E. French	Albert Buenger	C. A. Booth
F. E. Giesecke	S. H. Downs	E. K. Campbell
L. A. Harding	H. N. Kitchell	John Howatt
G. L. Larson	H. R. Linn	A. J. Nesbitt
A. P. Kratz	Perry West	J. H. Walker

Executive Committee

John Howatt, *Chairman*

F. E. Giesecke H. R. Linn

COMMITTEE ON RESEARCH (*continued*)

Finance Committee

A. J. Nesbitt, *Chairman*

N. D. Adams | E. C. Evans | G. L. Larson | J. H. Walker

Technical Advisory Committees

Air Conditions and Their Relation to Living Comfort: C. P. Yaglou, *Chairman;* J. J. Aeberly, W. L. Fleisher, D. E. French, Dr. R. R. Sayers and Dr. C.-E. A. Winslow.

Air Conditioning in Treatment of Diseases: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, Margaret Ingels, H. R. Linn and E. L. Stammer.

Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke): H. C. Murphy, *Chairman;* J. J. Bloomfield, M. I. Dorfan, Philip Drinker, Dr. Leonard Greenburg, S. R. Lewis, T. W. Pangborn, F. B. Rowley and Games Slayter.

Correlating Thermal Research: R. M. Conner, *Chairman;* D. S. Boyden, J. C. Fitts, H. T. Richardson and Perry West.

Corrosion: J. H. Walker, *Chairman;* H. F. Bain, E. L. Chappell, W. H. Driscoll and R. R. Seeber.

Direct and Indirect Radiation with Gravity Air Circulation: H. F. Hutzel, *Chairman;* A. P. Kratz, H. R. Linn, J. F. McIntire, J. P. Magos, T. A. Novotney, R. N. Trane and G. L. Tuve.

Gas Heating Equipment: W. E. Stark, *Chairman;* Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, E. L. Tornquist and H. L. Whitelaw.

Heat Requirements of Buildings: D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, E. C. Rack, F. B. Rowley, R. J. J. Tennant and J. H. Walker.

Heat Transfer of Finned Tubes with Forced Air Circulation: F. B. Rowley, *Chairman;* H. F. Bain, H. F. Hutzel, W. G. King, A. P. Kratz, E. J. Lindseth, L. P. Saunders, G. L. Tuve and W. E. Stark.

Minimum Temperature and Method of Introduction of Cooling Air in Classrooms: Perry West, *Chairman;* J. D. Cassell, S. R. Lewis, J. R. McColl, A. J. Nesbitt, G. E. Otis and C.-E. A. Winslow.

Oil Burning Devices: H. F. Tapp, *Chairman;* Elliott Harrington, F. B. Howell, J. H. McIlvaine, L. E. Seeley and T. H. Smoot.

Pipe and Tubing (Sizes) Carrying Low Pressure Steam or Hot Water: S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, R. R. Seeber and W. K. Simpson.

Refrigeration in Relation to Air Treatment: A. P. Kratz, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick and J. H. Walker.

Sound in Relation to Heating and Ventilation: V. O. Knudsen, *Chairman;* Carl Ashley, C. A. Booth, F. C. McIntosh, R. F. Norris, J. S. Parkinson, C. H. Randolph, J. P. Reis and G. T. Stanton.

Ventilation of Garages and Bus Terminals: E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore and A. H. Sluss.

Cooperating Committee

A. S. H. V. E. representative on *National Research Council;* Prof. F. E. Giesecke (3 years).

SPECIAL COMMITTEES

Committee on Admission and Advancement: A. J. Offner, *Chairman* (*two years*); J. G. Eadie (*one year*), and E. N. Sanbern (*three years*).

Publication Committee: W. M. Sawdon, *Chairman;* A. P. Kratz and M. C. Beman.

Committee on Constitution and By-Laws: Thornton Lewis, *Chairman;* W. T. Jones and O. W. Ott.

Guide Publication Committee: W. L. Fleisher, *Chairman;* John Howatt, G. L. Larson, S. R. Lewis, E. N. McDonnell, W. M. Sawdon and J. H. Walker. J. L. Blackshaw, *technical assistant.*

Committee on Ventilation Standards: W. H. Driscoll, *Chairman;* J. J. Aeberly, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard.

Committee on Code for Testing and Rating Condensation and Vacuum Pumps: John Howatt, *Chairman;* W. H. Driscoll, L. A. Harding and F. J. Linsenmeyer.

Committee on Code for Testing and Rating Convectors: R. N. Trane, *Chairman;* E. H. Beling, R. F. Connell, M. Dillman, W. Ewald, J. H. Holbrook, Hugo Hutzel, A. P. Kratz, M. G. Steele and O. G. Wendel.

Nominating Committee for 1934

Chapters	*Representative*	*Alternate*
Cincinnati	J. J. Braun	H. E. Sproull
Cleveland	C. F. Eveleth	F. A. Kitchen
Illinois	J. J. Aeberly	J. J. Hayes
Kansas City	W. A. Russell	E. K. Campbell
Massachusetts	J. F. Tuttle	W. F. Gilling
Michigan	W. G. Boales	Tom Brown
Western Michigan	S. H. Downs	J. H. Van Alsburg
Minnesota	N. D. Adams	C. E. Lewis
New York	H. W. Fiedler	V. J. Cucci
Western New York	D. J. Mahoney	J. J. Yager
Ontario	H. S. Moore	F. E. Ellis
Pacific Northwest	A. L. Pollard	Lincoln Bouillon
Philadelphia	W. R. Eichberg	M. F. Blankin
Pittsburgh	F. C. McIntosh	F. C. Houghten
St. Louis	R. J. Tenkonohy	
Southern California	L. H. Polderman	R. L. Gifford
Wisconsin	E. A. Jones	Ernest Szekely

Officers of Local Chapters—1934

Cincinnati

Headquarters, Cincinnati
Meets: Second Tuesday in Month

President, H. N. Kitchell
Vice-President, G. B. Houliston
Secretary-Treasurer, E. B. Royer
Board of Governors: C. J. Kiefer, K. A. Wright and the officers

Cleveland

Headquarters, Cleveland
Meets: Second Thursday in Month

President, M. F. Rather
Vice-President, H. B. Matzen
Secretary, E. J. Vermere
Treasurer, H. M. Nobis
Board of Governors: G. L. Tuve, C. F. Eveleth, F. A. Kitchen

Illinois

Headquarters, Chicago
Meets: Second Monday in Month

President, C. W. DeLand
Vice-President, R. E. Hattis
Secretary, J. J. Hayes
Treasurer, J. H. Milliken
Board of Governors: G. H. Blanding, J. J. Haines, S. I. Rottmayer, H. R. Linn

Kansas City

Headquarters, Kansas City, Mo.
Meets: Second Monday in Month

President, L. A. Stephenson
Vice-President, C. A. Weiss
Secretary, L. R. Chase
Treasurer, G. L. Bliss
Board of Governors: David Caleb, C. A. Flarsheim, Harry Filkins

Massachusetts

Headquarters, Boston
Meets: Second Monday in Month

President, R. S. Franklin
Vice-President, J. W. Brinton
Secretary, W. A. McPherson
Treasurer, E. W. Berchtold
Board of Governors: Leslie Clough, L. A. Brissette, G. B. Gerrish, C. T. Flint, A. B. Jackson and the officers

Michigan

Headquarters, Detroit
Meets: First Monday After 10th of Month

President, G. D. Winans
Vice-President, A. C. Wallich
Secretary, W. F. Arnoldy
Treasurer, R. K. Milward
Board of Governors: Edward Glanz, J. S. Kilner, H. E. Paetz, F. J. Feely

Western Michigan

Headquarters, Grand Rapids
Meets: Second Monday in Month

President, S. H. Downs
Vice-President, P. O. Wierenga
Secretary, W. G. Schlichting
Treasurer, W. J. Temple
Board of Governors: K. L. Ziesse, H. F. Reid, Harry Christenson

Minnesota

Headquarters, Minneapolis
Meets: Second Monday in Month

President, C. E. Lewis
Vice-President, C. E. Gausman
Secretary-Treasurer, R. E. Backstrom
Board of Governors: A. B. Algren, F. C. Winterer

New York

Headquarters, New York
Meets: Third Monday in Month

President, H. W. Fiedler
Vice-President, W. W. Timmis
Secretary, T. W. Reynolds
Treasurer, W. M. Heebner
Board of Governors: A. F. Hinrichsen, H. L. Alt, S. A. S. Patorno

Western New York

Headquarters, Buffalo
Meets: Second Monday in Month

President, J. J. Yager
Vice-President, W. E. Voisinet
Secretary, P. S. Hedley
Treasurer, B. C. Candee
Board of Governors: R. T. Coe, L. A. Harding, C. W. Farrar, W. G. Fraser, Roswell Farnham, M. S. Jackson, C. A. Evans, O. K. Dyer, Joseph Davis, C. M. Beman, D. J. Mahoney

Ontario

Headquarters, Toronto, Ont.
Meets: First Monday Every Other Month

President, W. R. Blackhall
Vice-President, M. W. Shears
Secretary-Treasurer, H. R. Roth
Board of Governors: J. S. Paterson, Thomas McDonald, G. A. Playfair, W. P. Boddington

Pacific Northwest

Headquarters, Seattle
Meets: Second Tuesday in Month

President, A. L. Pollard
Vice-President, Lincoln Bouillon
Secretary, S. D. Peterson
Treasurer, W. W. Cox
Board of Governors: P. M. O'Connell, C. F. Twist, W. E. Beggs

Philadelphia

Headquarters, Philadelphia
Meets: Second Thursday in Month

President, W. P. Culbert
Vice-President, J. H. Hucker
Secretary, W. R. Eichberg
Treasurer, W. F. Smith
Board of Governors: M. F. Blankin, A. C. Caldwell, Alexander McClintock, Jr.

Pittsburgh

Headquarters, Pittsburgh
Meets: Second Monday in Month

President, P. A. Edwards
Vice-President, L. B. Pittock
Secretary, J. L. Blackshaw
Treasurer, H. A. Beighel
Board of Governors: F. H. Hecht, G. S. McElroy, R. J. J. Tennant

St. Louis

Headquarters, St. Louis
Meets: First Wednesday in Month

President, Paul Sodemann
First Vice-President, J. W. Cooper
Second Vice-President, C. R. Davis
Secretary, A. L. Walters
Treasurer, G. W. F. Myers
Board of Governors: C. A. Pickett, J. M. Foster, E. E. Carlson, E. B. Evleth

Southern California

Headquarters, Los Angeles
Meets: Second Tuesday in Month

President, W. H. C. Ness
Vice-President, W. E. Barnum, Jr.
Secretary, E. H. Kendall
Treasurer, H. L. Warren
Board of Governors: L. H. Polderman, W. E. Cranston, H. H. Douglas, A. H. Simonds

Wisconsin

Headquarters, Milwaukee
Meets: Third Monday in Month

President, John S. Jung
Vice-President, Ernest Szekely
Secretary, G. E. Hochstein
Treasurer, Martin Erickson
Board of Governors: G. L. Larson, E. A. Jones, V. A. Berghoefer

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 968

FORTIETH ANNUAL MEETING, 1934

WITH 650 members, guests and ladies in attendance, the discussion of 20 papers at the six technical sessions and the adoption of the Code for Testing Vacuum Pumps, the 40th Annual Meeting of the Society at Hotel Biltmore from February 5 to 9 was pronounced an outstanding success and a tribute to the New York Chapter members who acted as hosts for the occasion.

The week of February 5 was featured by a splendid Exposition of heating, ventilating and air conditioning equipment at Grand Central Palace, with an audience of 54,000 people during the five days and the meeting of numerous groups of manufacturers. On February 9 a conference to discuss the advisability of creating a basic code for manufacturers of heating, ventilating and air conditioning equipment was considered at a meeting of 200 manufacturers' representatives and code authorities, who were addressed by Beverly S. King, Assistant Deputy Administrator, and Wm. J. Hoff of the Compliance Division of NRA.

Pres. W. T. Jones called the 40th Annual Meeting of the Society to order in the Ballroom of the Hotel Biltmore, New York. He announced that on January 26, in accordance with the requirement of the Constitution and By-Laws, a meeting was opened by Vice-Pres. C. V. Haynes in the Hotel Biltmore, but there being no quorum present, adjournment was taken until February 5.

Arthur Ritter, chairman of the Committee on Arrangements, greeted the members present in behalf of the New York Chapter and President Jones responded briefly.

President Jones then read his report.

Report of President

Since the last Annual Meeting of the Society, I have visited all of the seventeen Chapters, and have met with two groups of members in cities where we do not have Chapters, namely Washington, D. C., and San Francisco, Calif. At all of these meetings I have tried to tell the members about the Status Quo of the Society, and it is very pleasing to me to report that throughout our organization I have found a wholesome interest in the affairs of the Society, and a healthy enthusiasm in support of Society activities. The time is not far distant when Chapters can be established in San Francisco and Washington.

The past year has not been notable for its constructive achievement, but rather, your Officers have pursued a course of planned retrenchment, in an endeavor to carry out a pay as we go policy on a drastically reduced income. Other reports will give the details of membership and finances, and it will be noted that the Society is in a healthy condition, although naturally, some drop in membership is bound to occur in times such as we are passing through.

The reduction of dues to $18.00 combined with an active effort on the part of our members has resulted in a pleasing number of new applications, and I urge continued support of our Membership Committee's efforts so that we may regain not only the former number of members, but materially increase it.

This Society is unique in that it returns to each member more in the form of tangible value than he pays in for dues. This condition is possible due to the willingness of a considerable number of our members to contribute liberally of their time, money and ability to Society activities, and also due to the farsighted wisdom of some of the splendid men who have served the Society in the past.

This condition can not continue indefinitely on our present basis of membership, and neither should our members expect it to do so. In these unusual times it is proper to expect each Society employee to do a little more and to expect our membership to contribute time and ability for the good of the cause, but when business conditions again approach normal, the Society may be expected to pay for some of the services now being contributed. That is the day we must look forward to and plan for, and I venture to direct your thoughts along these lines.

According to our By-Laws, on next January first the admission fee of $15.00 is to be restored and our dues are to be $25.00, as formerly. Unless conditions change to an extent not now anticipated, it is my opinion that it will not be practicable to return to the former basis on January 1, 1935, and the present or similar rate of dues will have to be continued for a while longer.

It is not my purpose to inject this troublesome question of amount of dues into this report except in so far as this problem affects the point I have in mind for your consideration.

On the present basis of $18.00 per year dues and with 40% allocated to research, Society activities have to be conducted on $10.80 dues from each member. Obviously this amount is too small for a Society of only about 1,900 members to operate on with our present rather ambitious program, and we shall either have to curtail our activities or else obtain more members. In looking over the field and in talking with members of the Society in various parts of the country, I am convinced that there are at least 2,000 men in the industry who are well qualified for membership who do not now belong to the Society. I would suggest that the biggest problem before this Society is to get these men to join our membership and take their share of responsibility in carrying on the work the Society is doing. This increase in our membership would not only make the Society a more powerful organization, but would enable the Research Laboratory to operate an extensive program without the necessity of soliciting funds outside the Society. It would also mean that the Society could be operated at or near the present scale of dues.

The Membership Committee alone can accomplish this desirable end; the program will require the backing of the entire Society membership. If all our members knew as much about the Society as we who are here do, the task would not be difficult. The problem is worthy the consideration of the Advisory Council.

We frequently hear the statement that Society membership consists principally of

salesmen, with a few practicing consulting engineers scattered throughout the country. An analysis of our present membership shows the following divisions by occupations:

Consulting Engineers, Architects and Engineers in their employ	11.5%
Heating, Ventilating Contractors and their Estimators and Engineers	18 %
Sales Engineers	19.4%
College Professors	2.2%
Government and Municipal Engineers	2.5%
Research Engineers	4 %
District Heating and Utilities Engineers	2.5%
Manufacturing Executives	10.8%
District and Branch Managers of Manufacturers	8.5%
Association and Publicity Engineers	2.1%
Students	7 %
Unclassified	11.5%

I am giving this information to correct the misapprehension in the minds of some people regarding the occupations followed by our membership.

The Society is indeed fortunate in having an exceptionally fine group of men on the Council this year. No President has ever been blessed with a better Council than I have had. This Council has had to solve some difficult problems most of which are due to a materially reduced income. A considerable number of our members have found it impossible to pay their dues, which fact combined with the comparatively large reduction in the amount of dues has presented some financial problems which have not been easily solved. Your Finance Committee is to be highly commended for an exceptionally fine piece of work.

In my visits to the various Chapters, I have found that the Society activity in which the members take the most interest is the Research program. It has been a particularly difficult year for the Committee on Research, and this Committee has functioned in a highly creditable manner. Their problem has been similar to that of the Society in that the activities have to be conducted on a budget about 50 per cent of what has been considered normal.

When business conditions are good and Society finances are not a great problem, it is easy to find ways of extending our activities, but when times change and Society finances suffer, it is not so easy to curtail our activities to meet the reduced budget, without sacrificing some activities which past Officers of the Society have achieved. The Society today is in a sound financial condition, due largely to the wise planning of some of the Officers in years gone by. Our structure is built upon a firm foundation, due to the foresight of these men, and they are entitled to our gratitude and appreciation for the heritage they have given us in this Society.

Without vision, without foresight, organizations like ours must inevitably perish. Without worthy conceptions animating our ideals we would drift from easy complacency in our brief years of prosperity, to disorganized despair in the long years of hardship. Let us, therefore, always keep before us the fine ideals of this Society; let us all work to the common end of making this industry a better industry because of our efforts; let us prove ourselves worthy of the great age in which we live, and unite in this Society to strive for constantly higher standards and higher ideals.

Respectfully submitted,

W. T. Jones, *President.*

On motion of H. G. Issertell, seconded by J. D. Cassell, it was voted that the Report of the President be placed in the minutes of the meeting.

President Jones called upon the Secretary to read the Report of the Council.

Report of Council

The Council met in Cincinnati, Detroit, Chicago and New York since the last Annual Meeting and because of the small number of meetings much work was carried

on through correspondence. The business of the Society was administered by the Council and the group of technical committees appointed by Pres. W. T. Jones were responsible for the meeting programs, production of THE GUIDE and the Code for Testing and Rating Condensation and Vacuum Pumps. The work of other special committees passed on the qualifications of candidates for membership, proposed revisions to the Constitution and By-Laws, reviewed papers offered for publication and aided in strengthening the Society's membership.

In January, 1933, the Council confirmed the appointment of four Council Committees suggested by President Jones and voted the appointment of A. V. Hutchinson as Secretary, selected depositories for Society Funds and adopted the budget for 1933 with an estimated income of $54,924 and an expense of $54,100.

The invitation of the Michigan Chapter to hold the Semi-Annual Meeting, 1933, in Detroit and the invitation of the New York Chapter to have the 40th Annual Meeting, 1934, at New York coincident with the Third International Heating and Ventilating Exposition, were accepted. The petition of students at Carnegie Institute of Technology for a Student Chapter was granted and the application of the New York Chapter to institute the plan of Limited Chapter Membership was approved.

In accordance with the provisions of the By-Laws the Council nominated members to serve on the Committee on Research for a three-year term beginning in 1934 as follows: C. A. Booth, E. K. Campbell, John Howatt, A. J. Nesbitt and J. H. Walker.

New rates for catalog data space in THE GUIDE, 1934, proposed by the Guide Publication Committee and recommended by the Finance Committee were approved and price of THE GUIDE, 1934, for sale to non-members of the Society was set at $5 for single copies.

Exhibits were maintained at the *American Oil Burner Association* meeting, in Chicago, during June with the cooperation of the Illinois Chapter and an exhibit at the meeting of the *American Society for Testing Materials* in Chicago, during the same month, featured the research work of the Society and was in charge of Director Houghten.

In August the Council voted to approve in principle the Code of Fair Competition for the Professional Engineer Division of the Construction Industry and appointed John G. Eadie, consulting engineer of New York, as the Society's representative on the National Control Committee.

In accordance with the By-Law provision the Council authorized the employment of a C. P. A. to audit the Society's Books for 1933 and as required by the By-Laws, set the dues rate for 1934 for Members and Associates at $18, Juniors $10 and Students $3 without Initiation Fees.

Changes in the editorial contract were authorized to provide for any revision in the subscription rate and permission was granted for an assignment of the contract to the Keeney Publishing Co. by Domestic Engineering Co., successors to Engineering Publications, Inc.

The Election and reinstatement of 256 members were confirmed by letter ballot and the routine motions on resignations and cancellations of 411 memberships were adopted. The present status is 1887.

Because of the unauthorized use of copyright reports of the Society, in sales promotion literature and in recently published books, the Council made vigorous protest to those responsible. A proposal for commercial testing was received and careful study of the plan indicated that the idea could not be accepted as it was not in accord with the Society's Constitutions and By-Laws and the Regulations of the Committee on Research.

The program for the Annual Meeting was approved and an exhibit at the Heating and Ventilating Exposition was authorized.

The several Council Committees devoted much time and effort to the business of the Society and this brief report is submitted for consideration of the membership in the belief that the work of the Society's governing body will be of interest.

THE COUNCIL.

On motion of F. D. Mensing, properly seconded, it was voted that the Report of the Council be received and filed.

F. D. MENSING: The Council was put in the position here lately of having to support the President through the National Recovery Act. The Society was asked to support the movement for a Code for Engineers. The Council did support the President, in which I backed them up. On the other hand, I question whether the Council would have supported any other code for anybody. Our constitution precludes the possibility of supporting any type of trade approval. Am I correct, Mr. Secretary?

A. V. HUTCHINSON: I think that is correct.

MR. MENSING: The National Recovery Act, as I understand it, comes under that category. Our Constitution states that unless the proceedings of the Council are questioned at the next meeting of the Society, they are not subject to being questioned again.

So I bring up this point as a matter of record and make the suggestion that the constitution be amended to give the Council more authority.

President Jones asked the Secretary to read his report.

Report of Secretary

Under a drastically curtailed budget the activities of the Society in 1933 presented a serious administrative problem. It was necessary to carry on to completion many projects under way prior to the beginning of the year and added to this were increased demands for services of various kinds, including employment, NRA code information and an unusual increase in membership applications. A new application form was prepared and distributed in order to simplify the procedure of election under the provisions of the new Constitution and By-Laws. Copies of the Constitution and By-Laws adopted at the last Annual Meeting were printed and distributed to all members.

Three Codes were completed since the last Annual Meeting and were sent to members, namely, the Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section); Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work and the Code for Testing and Rating Return Line Vacuum Heating Pumps. The vote-by-letter ballot reported by the tellers is Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)—For, 170; Against, 5. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work—For, 175; Against, 6. Code for Testing and Rating Return Line Vacuum Pumps—For, 180; Against, 1.

Production of THE GUIDE, 1933, required an unusually large amount of time of the headquarters staff because of the size of this volume which presents the largest text section ever compiled.

Under the direction of W. L. Fleisher, 50 men were directly engaged in the preparation of the manuscript for the text. The response of manufacturers was excellent so that the catalog data section is slightly larger than in last year's edition.

Two volumes of TRANSACTIONS are nearly ready for distribution to the membership and as these are two of the largest volumes produced in recent years, they imposed a heavy editorial burden on the headquarters staff as it was necessary to edit manuscript and discussions to meet the budget allowance. It is expected that copies of these two books will be mailed to members who paid dues for the years 1931 and 1932 about March 1.

A factor that stimulated interest in Society activities was President Jones' visits to Chapters. He told the story of the Society and what membership in the organization means so that local membership chairmen reported that the information given by Mr. Jones was very helpful in interesting candidates for membership. Chapters in 17 cities were visited by President Jones. During the year 256 men were elected

to membership and the present status is 1,887, consisting of 1,238 Members, 363 Associates, 131 Juniors, 125 Students, 28 Life Members, 2 Honorary Members. Two unusually well-attended meetings were held during the past year, the 39th Annual Meeting in Cincinnati where the Local Chapter acted as hosts, and the Semi-Annual Meeting in Detroit where the Michigan Chapter Members accepted the responsibility for the entertainment of the Society's members and guests. The Committees who were responsible for arranging these meetings worked very hard in making them successful and the Society is indebted for the cooperation of each and every one who served.

Due to the size of the meetings held during the year the volume of publication work was exceptionally large and after suitable subjects had been suggested by the Program Committee, papers were obtained and then were reviewed by the Publication Committee before appearing in the Journal of the Society. Each month the chapter activities were reported through the cooperation of the local secretaries. The headquarters office handled the routine correspondence, recording of minutes of meetings and other duties required by the Constitution and By-Laws.

All of the work has been carried on with a personnel of five people and I am glad to publicly express to the members of the staff my thanks for the extra effort that they have given and the many hours that they have spent without additional compensation in carrying on the work for the benefit of the Society membership.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

The Report of the Tellers of Election was given by M. E. Durkee:

Report of Tellers

Your Board of Tellers wishes to report results of the vote for election of Officers and Council and members of the Committee on Research to serve in 1934:

President—C. V. HAYNES	372
First Vice-President—JOHN HOWATT	375
Second Vice-President—G. L. LARSON	373
Treasurer—D. S. BOYDEN	374

Members of the Council—Three Year Term:

M. C. BEMAN	374
E. H. GURNEY	375
O. W. OTT	374
W. A. RUSSELL	374

Members of the Committee on Research—Three Year Term:

C. A. BOOTH	375
E. K. CAMPBELL	375
JOHN HOWATT	375
A. J. NESBITT	375
J. H. WALKER	375
Total ballots cast	445
Disqualified	70
Qualified ballots	375

TELLERS OF ELECTION

W. R. EICHBERG, *Chairman,*
G. A. DORNHEIM
M. E. DURKEE

At the second session, Tuesday, February 6, 9:30 a. m., President Jones called the meeting to order and introduced John Howatt, Chairman of the Finance Committee, who reported on the condition of the Society.

Report of Finance Committee

If a budget is described as a device whereby the worrying is done before the money is spent rather than after, the description would be a true one if the expenditures only had to be budgeted, but we must prepare a budget of anticipated revenue as well as anticipated expenditures. Therefore, our financial worries of 1933 did not cease with the approval of our budget a year ago, because the anticipated and budgeted revenues were not being obtained. There are three main sources of income for the Society.

Income from the Contract with H. P. A. C.

Income from Guide Advertising and Sales.

Income from Members.

Budgeted revenues were obtained from the H. P. A. C. contract and from THE GUIDE but were not received from members.

The gross revenue to the society fell below our estimates to the extent of $5,047.77. The expenses of the Society were $1,987.54 below the budget, leaving a gap of $3,060.23 between budgeted income and expense. However, the budget provided an apparent surplus or cushion of $2,324.00, leaving a net loss for the year of $736.23.

There are accounts receivable from members dues prior to 1933 totaling $9,617.24 and for 1933 totaling $11,652.00. In setting up the assets of the society as of January 1, 1934, all claims for members dues prior to 1933 have been considered of no value, and 75 per cent of the unpaid dues of 1933 have been considered uncollectible. Therefore, of an unpaid dues account of $21,269.24 but $2,913.00 has been considered collectible and has been set up as revenue for 1934.

A year ago your Finance Committee was authorized to sell $5,000.00 par value bonds of the Wisconsin Power and Light to provide for the salary of a man retained to work on GUIDE sales and promotion. Because of the low price market prevailing throughout 1933 it was considered unwise to sell bonds of this class at the price obtainable, so they are still held in the reserve funds of the Society.

Securities owned by the Society exclusive of Research Funds as of December 31, 1933, cost $26,781.25 with a market value of $13,042.00, a depreciation of approximately 50 per cent. Since the beginning of the year, however, the market value of these securities has increased nearly $3,000 and it is confidently expected the market for them will improve considerably in 1934. The monies received from the sale of C. S. P. & L. debentures has been invested in U. S. Government 3⅛ bonds.

The capital account or net worth of the society on January 1, 1933, was $32,823.93. The capital account or net worth of the society on December 31, 1933, was $32,458.10, the shrinkage being due to the loss on the sale of C. S. bonds.

The balance sheet at the beginning of this year shows:

$11,939.99 cash on hand.
$21,887.90 collectible accounts.
$30,721.38 invested assets (at cost).
$4,866.79 other assets and inventories.
Only $455.36 in notes receivable from members for past dues.
$36,781.48 current assets.

And total assets of $69,760.42; against these there are current liabilities of $15,372.77. The financial affairs of the Society are in sound condition with current assets 2.3 times current liabilities.

FINANCE COMMITTEE
JOHN HOWATT, *Chairman*

Report of Certified Public Accountant

January 15, 1934.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
51 MADISON AVENUE,
NEW YORK, NEW YORK.

Gentlemen:

We have made an examination of your books and accounts for the calendar year ended December 31, 1933, and are submitting the attached Statements and Schedules as our Report thereon.

The Accounts Receivable shown on Statement "A" include amounts due from Members on account of dues in the amount of $21,269.24. We have consulted with your Secretary, Mr. A. V. Hutchinson, regarding the probable collectibility of these accounts and have accordingly considered seventy-five per cent of the unpaid 1933 dues and the entire amount of unpaid dues for prior years as uncollectible, the details of which are as follows:

DUES RECEIVABLE		
1933 Accounts		$11,652.00
Prior Years Accounts		9,617.24
		$21,269.24
DEDUCT		
Reserve for Uncollectible Dues 1933 Accounts (75%)	$8,739.00	
Prior Years Accounts (All)	9,617.24	18,356.24
COLLECTIBLE DUES		$2,913.00

It will be seen that the reserve provided for uncollectible accounts in the amount of $18,356.24 is quite large but we believe this is justified in view of prior years collections. The amount due from advertisements as shown on Statement "A" includes the sum of $15,213.50 receivable from advertisers in the 1934 GUIDE, all of which has been considered collectible. Reserves set up for other advertising and sundry accounts receivable are in accordance with recommendations made by Mr. A. V. Hutchinson, which, in our opinion, appears to be adequate to provide for any future losses from this source.

The inventory of 1926-1930 transactions, emblems and certificate frames and office supplies was furnished us by your office staff, and an analysis of the records indicates that this is correct. Invested Assets as shown consist in the main of securities owned, the cost to the Society of which is greatly in excess of their market value as at December 31, 1933. Other than showing these market values (See Schedule "3") no attempt has been made to reconcile these apparent differences on our report. We have included in invested assets the sum of $675.00 representing the income on the securities owned in the Society's Endowment Fund. We understand it has been the past policy of the Society to transfer this income to the Research Laboratory but this has not been done in 1933, in the absence of any specific recommendations to this effect by the Council. Again, the receipt of $1,842.38 from the sale of five Cities Service Bonds belonging to the Society's Endowment Fund was deposited in the Society's Savings Account in the Bowery Savings Bank, New York City. We understand that this money is not available for use by the Society and therefore, have set this amount aside under the caption "Invested Assets Endowment Fund" on the Balance Sheet.

In accordance with past custom the furniture and fixtures were depreciated at the rate of 10 per cent per annum whereas no depreciation was taken on the Library. Under the caption "Current Liabilities" on the Balance Sheet as shown, is the sum

of $3,020.70 representing a reserve set aside for completion of the 1931-1932 transactions. This consists of

Reserves set up by Council for	
1931 Transactions	$3,500.00
1932 Transactions	3,000.00
	$6,500.00
LESS	
Expenses Incurred to date account of the above	3,479.30
BALANCE, TO COMPLETE	$3,020.70

The details of other reserves provided for, that is, $5,000.00 to complete the 1934 GUIDE, $850.00 for expenses of mailing the 1934 GUIDE (Controlled Distribution) and $3,000.00 which is the estimated cost of compiling the 1933 transactions, have been furnished by your Secretary.

The Balance Sheet also includes the sum of $504.16 due the Research Laboratory, the details of which are as follows:

Balance due to Research Laboratory January 1, 1933		$2,834.94
ADD — 40% of 1933 Members' Dues (Statement "B")		
Renewals	$5,610.82	
New Members	592.20	6,203.02
Interest on Securities Owned by Research Laboratory Deposited in Society Funds		135.00
Cancellation of 1932 check shown under caption "Cash on Hand for Deposit" on December 31, 1932 Balance Sheet		820.00
Difference between 1933 Dues Paid and Actual Amount Due		49.00
		$10,041.96
LESS Cash Paid to Research Laboratory in 1932		9,537.80
Balance Due Research Laboratory Dec. 31, 1933		$ 504.16

In arriving at the total amount of income from Members' Dues for 1933 on Statement "B" of this Report, 40 per cent of the Dues charged to Members and Associates was reserved for the Research Laboratory which is in accordance with Sections 5 and 7, Article B-4, of the By-Laws. In preparing this Statement we believed that it would be interesting to have a comparison of the 1933 and 1932 operations and therefore Statement "B" has been compiled in comparative form. It will be seen that a loss of $736.23 was incurred on operations for the year 1933 despite the fact that expenses were drastically curtailed. Nevertheless, this compares favorably with a loss of $2,356.53 incurred for the similar period in 1932. The Statement would indicate that the 1933 loss was incurred in the main from the reduced income from Members' dues. This may be explained by the action of the Council in reducing the yearly dues and waiving the initiation fees, the effect of which was felt in the year 1933. Furthermore, the Society's membership decreased in 1933. (See Schedule "7".) Again, it will be seen that a proportionately greater amount has been provided for uncollectible dues in 1933 than in 1932, which has the effect of reducing the dues income for 1933.

The loss of $629.60 incurred on the sale of one Cities Service Bond belonging to the free invested funds of the Society has not been considered as an operating loss for the period and therefore is not reflected in the Society's $736.23 loss on operations in 1933, as stated above.

In accordance with previous reports, we have prepared a Statement of Budget Revenues Anticipated and Actual Receipts and Budget Appropriations and Actual Disbursements, the details of which are given on Statement "C" and which, we believe, is self-explanatory.

In preparing the Balance Sheet we have deviated from previous reports in that we have compiled separate Balance Sheets of the Society and the Research Laboratory, which we believe makes for greater clarity.

Statement "D" shows these separate Assets and Liabilities of the Research Laboratory as of December 31, 1933.

In conformity with Statement "B" prepared by us upon the operations of the Society, we have compiled the Research Laboratory Statement of Income and Profit and Loss in comparative form. This indicates that a loss of $5,935.51 was incurred by the Laboratory in 1933, whereas the loss in the similar period in 1930 amounted to $2,288.71. This latter figure differs from the 1932 profit of $1,567.54 shown in the 1932 report by $3,856.25, which is the difference between the amount of $16,795.92 shown as income from dues in the 1932 Research Laboratory report and $12,939.67 shown as income from 1932 dues on Statement "E" of this Report. The $16,795.92 amount represents the net cash transferred by the Society to the Research Laboratory in 1932 and not the dues income for 1932 as the 1932 report would indicate. Rather, in our opinion the Research Laboratory was entitled, among other things, to 40 per cent of the dues as shown in the 1932 Society's operating report, which is the figure shown, namely, $12,939.67. Statement "F" gives the results between the actual receipts and disbursements of the Research Laboratory for 1933 and the amounts of income and expenses as per the Budget for that period, which we believe is self-explanatory.

If any further data are desired we shall be pleased to furnish it from the working papers on file at our office.

Respectfully submitted,

WILLIAM A. MILLIGAN & COMPANY,

Certified Public Accountants.

BALANCE SHEET

As of December 31, 1933

ASSETS

CURRENT ASSETS				
Cash in Banks (as per Schedule "1")			$11,836.99	
Cash on Hand, New York Headquarters			103.00	$11,939.99
Accounts Receivable	*Total*	*Reserve for Bad and Doubtful Accounts*	*Collectible Accounts*	
Membership Dues	$21,269.24	$18,356.24	$ 2,913.00	
Advertisements	20,407.43	2,717.42	17,690.01	
Other Accounts Receivable	1,396.84	222.95	1,173.89	
TOTAL ACCOUNTS RECEIVABLE	$43,073.51			
DEDUCT: Reserve for Bad and Doubtful Accounts		$21,296.61		
COLLECTIBLE ACCOUNTS				$21,776.90
NOTES RECEIVABLE				455.36
INVENTORIES (At cost)				
1926–1930 Transactions			$ 2,203.83	
Emblems and Certificate Frames			30.40	
Postage, Printing and Supplies			375.00	2,609.23
TOTAL CURRENT ASSETS				$36,781.48
INVESTED ASSETS (as per Schedule "2")				
Free Funds			$12,007.08	
Restricted Funds—Endowments			18,714.30	
TOTAL INVESTED ASSETS (at cost)				30,721.38
Furniture and Fixtures		$ 5,215.46		
DEDUCT—Reserve for Depreciation		3,257.90	$ 1,957.56	
Library			300.00	
TOTAL FIXED ASSETS				2,257.56
				$69,760.42

STATEMENT "A"

LIABILITIES AND CAPITAL

CURRENT LIABILITIES		
Sundry Accounts Payable (Includes part cost of Printing 1931–1932 Transactions and 1934 GUIDE)		$ 5,997.91
RESERVE FOR COMPLETION OF		
1931–1932 Transactions	$ 3,020.70	
1934 Guide	5,000.00	
Mailing 1934 GUIDE (Controlled Distribution)	850.00	8,870.70
Due Research Laboratory		504.16
TOTAL CURRENT LIABILITIES		$15,372.77
DEFERRED INCOME—PREPAID DUES		215.25
RESERVE FOR 1933 TRANSACTIONS		3,000.00
ENDOWMENTS		
Society	$17,644.46	
F. Paul Anderson Award	1,069.84	18,714.30
CAPITAL ACCOUNT, being net worth of Society, Dec. 31, 1933 (See Statement "B") (exclusive of Research Laboratory)		32,458.10
		$69,760.42

STATEMENT "B"

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS
For the Years Ended December 31, 1933 and December 31, 1932

YEAR 1932			INCOME			YEAR 1933
			MEMBERS' DUES			
	$50,525.00		RENEWALS—Members and Associates		$33,372.00	
			DEDUCT			
		$ 6,928.00	Cancellations	$ 6,507.00		
		12,657.38	Provision for Unpaid Dues	12,837.94		
	19,585.38				19,344.94	
	$30,939.62				$14,027.06	
	12,375.85		DEDUCT 40% to Research Laboratory		5,610.82	
	$18,563.77				$ 8,416.24	
	1,660.00		Renewals—Junior Members (Net)		1,000.00	
	100.00		Renewals—Student Members (Net)		120.00	
$20,323.77			TOTAL MEMBERS' DUES FROM RENEWALS			$ 9,536.24
			NEW MEMBERS' DUES			
	$ 1,409.55		Members and Associates		$ 1,480.50	
	563.82		DEDUCT 40% to Research Laboratory		592.20	
	$ 845.73				$ 888.30	
	76.95		Junior Members		185.00	
	112.50		Student Members		237.00	
1,035.18			TOTAL NEW MEMBERS' DUES			1,310.30
970.00			Initiation Fees			......
$22,328.95			TOTAL INCOME FROM MEMBERS' DUES			$10,846.54
			INCOME FROM EDITORIAL CONTRACT			
	$17,333.36		Income		$12,866.70	
11,127.36	6,206.00		DEDUCT Members' Subscriptions		3,698.84	9,167.86
			INCOME FROM MISCELLANEOUS SALES			
	COST	SALES PRICE		SALES PRICE	COST	
	$267.78	$388.86	Reprints and Books	$473.57	$315.41	
	......	386.25	Transactions	70.41	47.09	
	1,357.21		Transactions written off (1925)		157.71	
	722.44	46.47	Codes	......	......	
	110.12	104.00	Emblems and Certificate frames	39.15	54.17	
		$925.58	TOTAL SALES PRICE	$583.13		
	$2,457.55		TOTAL COST		$574.38	
1,531.97			GROSS PROFIT OR LOSS MISCELLANEOUS SALES			8.75
873.71			INCOME FROM INVESTMENTS			763.21
$32,798.05			GROSS INCOME (Forwarded)			$20,786.36

STATEMENT "B"
(Continued)

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS
For the Years Ended December 31, 1933 and December 31, 1932

YEAR 1932						YEAR 1933
$32,798.05			GROSS INCOME (Brought Forward)			$20,786.36
			GROSS PROFIT FROM GUIDES			
	$17,371.14		1934 GUIDE—Advertisements		$15,213.50	
	(1933)		COSTS			
		$1,553.27	Paper Costs	$1,338.39		
		8,834.15	Printing and Binding Costs to Date	2,925.00		
		......	Estimated to Complete	5,000.00		
		428.60	Engraving and Art Work	389.26		
		6,831.27	Editorial Service and Clerical Salaries	1,500.00		
		......	Mailing Expense (Controlled Distribution)	850.00		
	18,758.29	1,111.00	Sales Promotion Advertising 1933 GUIDE	1,257.05	13,259.70	
	$ 1,387.15		GROSS PROFIT—1934 GUIDE (Loss 1933) (Before Including GUIDE Sales)		$ 1,953.80	
(1932)	$12,317.26		1933 GUIDE—SALES		$ 9,248.86	
			COSTS			
		$2,679.62	Mailing & Postage	$ 1,934.39		
	5,817.18	3,137.56	Sales Promotion Copy Sales	1,624.35	3,558.74	
	$ 6,500.08		GROSS PROFIT 1933 GUIDE (Before deducting Production Costs)		$ 5,690.12	
			ADDITIONAL INCOME 1933 GUIDE ADVS.		428.59	
			DEDUCT		$ 8,072.51	
	$ 1,500.00		Provision for Bad Debts		1,460.94	
	$ 5,000.08					
3,612.93			TOTAL GROSS PROFIT FROM GUIDES			6,611.57
$36,410.98			TOTAL GROSS INCOME FROM ALL SOURCES			$27,397.93
			EXPENSES			
	$15,694.53		Salaries Paid Secretary and Office Staff (Schedule "4")		$13,571.00	
	1,704.81		Traveling Expense—Secretary		652.57	
	734.92		Traveling Expense—President		629.12	
	210.31		Emergency Fund—President		154.41	
	4,543.92		Rent and Light		2,676.45	
	......		Moving Expense		55.00	
	1,704.77		General and Office Expense		448.94	
	461.94		Stationery and Supplies		342.55	
	605.12		Telephone Expense		538.97	
	194.77		Telegraph Expense		171.01	
	262.52		Multigraphing		347.28	
	519.88		Deprn. on Furniture and Fixtures		521.55	
	400.00		Professional Services		200.00	
	......		Contribution to Amer. Stand. Assn.		50.00	
	......		Constitution and By-Laws Ptg. Exp.		192.81	
	836.32		General Printing Expense		443.62	
	3,500.00		Allowance 1933 Transactions		3,000.00	
	2,477.36		Postage		1,562.48	
	725.00		Year Book		510.40	
	2,993.76		Meetings and Exhibit Expense		807.67	
	1,000.00		Chapter Allowance		1,000.00	
	......		Filing System		50.00	
	23.03		Awards and Medals		50.00	
	174.55		Bank Charges		158.33	
$38,767.51			TOTAL EXPENSES (Forwarded)			$28,134.16

STATEMENT "B"
(Continued)

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS
For the Years Ended December 31, 1933 and December 31, 1932

YEAR 1932				YEAR 1933
$36,410.98	TOTAL GROSS PROFIT FROM ALL SOURCES (Brought Forward)			$27,397.93
38,767.51	TOTAL EXPENSES (Brought Forward)			28,134.16
$ 2,356.53	LOSS ON OPERATIONS FOR THE YEAR (Statement "C")			$ 736.23
	CAPITAL ACCOUNT OF SOCIETY JANUARY 1, 1933		$32,823.93	
	ADD—Adjustment of Reserve for Completion of 1931 and 1932 Transactions to amount as per letter to Members of Council dated July 28, 1933.			
	Provided for in Previous Years			
	1931 Transactions	$4,000.00		
	1932 Transactions	3,500.00		
		$7,500.00		
	As per letter to Members of Council			
	1931 Transactions	$3,500.00		
	1932 Transactions	3,000.00		
		$6,500.00		
	DIFFERENCE		1,000.00	
			$33,823.93	
	DEDUCT			
	Loss on Sale of One Cities Service Bond			
	Cost	$ 981.99		
	Sale Price	352.39	629.60	33,194.33
	CAPITAL ACCOUNT OF SOCIETY, DEC. 31, 1933 (See Statement "A")			$32,458.10

STATEMENT "D"

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
RESEARCH LABORATORY

BALANCE SHEET
As of December 31, 1933

ASSETS

CASH			
Bankers Trust, 42nd Street & 5th Ave., New York City		$ 138.33	
Forbes National Bank, Pittsburgh, Pa.		1,195.72	
On Hand, Pittsburgh, Pa.		7.60	$1,341.65
DUE FROM SOCIETY			504.16
INVESTED ASSETS			
Free Funds—3 Metropolitan Edison Co. 4½'s due March 1, 1968		$3,007.50	
Accrued Interest on Above		45.00	
		$3,052.50	
RESTRICTED FUNDS—Endowment			
Cash in Bank for Savings, 280 4th Avenue, New York City	$354.06		
Cash in Bank of United States, New York City (represents 45% of total deposit, 55% dividend having been paid in previous years).	274.43	628.49	
TOTAL INVESTED FUNDS			3,680.99
			$5,526.80

LIABILITIES

ENDOWMENT FUND	$ 628.49
CAPITAL ACCOUNT, being net worth of Research Laboratory December 31, 1933 (See Statement "E")	4,898.31
	$5,526.80

STATEMENT "E"

COMPARATIVE STATEMENT OF INCOME AND PROFIT AND LOSS (RESEARCH LABORATORY)

For the Years Ended December 31, 1933 and December 31, 1932

YEAR 1932				YEAR 1933
		INCOME		
	$12,375.85	Share of Dues from A.S.H.V.E. Renewals	$5,610.82	
$12,939.67	563.82	New Members	592.20	$6,203.02
		Receipts from Contributions (as per Sch. "5")		
		Special	$ 625.00	
10,765.00		General	3,529.00	4,154.00
135.00		Interest on Securities Owned		135.00
36.41		Interest on Bank Balances		8.25
....		Refund—American Society Testing Material Exhibit		108.14
5,012.76		Exposition (Net)		......
1,025.00		Interest on Endowment Fund		......
$29,913.84		GROSS INCOME		$10,608.41
		EXPENSES		
		Committee on Research Expense Traveling Committee		
	$ 887.75	and Laboratory Personnel	$ 401.92	
	800.00	Salary—Clerical Society Hgts	......	
	2,800.00	Salary—Technical Adviser	500.00	
	......	General Printing and Promotion Expense	70.30	
	500.00	Correlating Thermal Research Exp	500.00	
	$4,987.75	TOTAL EXPENSES—Committee on Research	$1,472.22	
		Laboratory Expense (Pittsburgh)		
	$12,449.16	Salaries (as per Sched. "6")	$8,491.70	
	571.71	Laboratory Supplies & Equipment	472.54	
	904.84	Office Supplies and Expense	344.63	
	67.25	Meetings	......	
	$13,992.96	TOTAL LABORATORY EXPENSES (Pittsburgh)	$9,308.87	
		Cooperative Research Expense		
	$ 6,750.00	University of Illinois	$3,225.00	
	1,750.00	University of Minnesota	875.00	
	695.86	Harvard University	1,153.02	
	1,450.00	Yale University	200.00	
	2,143.64	Miscellaneous	50.00	
	$12,789.50	TOTAL COOPERATIVE RESEARCH EXPENSE	$5,503.02	
		CONTINGENCY FUND EXPENSES		
	$ 432.34	Exposition Exhibit Expense	$ 259.81	
$ 2,202.55		TOTAL EXPENSES		16,543.92
$ 2,288.71		LOSS FOR THE YEAR (See Statement "F")		$ 5,935.51
		CAPITAL ACCOUNT JANUARY 1, 1933		
		Balance of Specific Funds Account	$7,998.88	
		Amount Due from Society 1/1/33	2,834.94	10,833.82
		CAPITAL ACCOUNT DECEMBER 31, 1933 (See Statement "D")		$ 4,898.31

On motion of Prof. G. L. Larson, seconded by Prof. A. C. Willard, the Report of the Finance Committee was approved and made a part of the minutes of the meeting.

Report of the Committee on Research

The Report of the Committee on Research by G. L. Larson, Chairman, and F. C. Houghten, Director of the A. S. H. V. E. Research Laboratory, was presented by Professor Larson.

The Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held two meetings in 1933, one last January in Cincinnati, and one in June in Detroit. Important items of business, including readjustment of the budget, were cared for by the Chairman of the Committee on Research, the Chairman of the Research Finance Committee, the Technical Adviser to the Committee, and the Director of the Research Laboratory at a meeting in Chicago in September. Additional business of the Committee was handled by correspondence.

Financial difficulties attending the continuation of the depression made it necessary to further curtail all activities in order to keep expenditures within funds available. A Balance Sheet and a Statement of Income and Expense prepared by the C.P.A. is shown on p. 10.

1933 Research Contributors

National Warm Air Heating and Air Conditioning Association
Utilities Research Commission, Inc.
Aerofin Corp.
American Air Filter Co., Inc.
American Blower Corp.
American Radiator Co.
Barber-Colman Co.
Barnes & Jones
Brillo Manufacturing Co., Inc.
Buffalo Forge Co.
Bush Manufacturing Co.
Carrier Engineering Corp.
Detroit Edison Co.
Detroit Stoker Co.
Donohoe Building Equipment Co.
C. A. Dunham Co.
Frigidaire Corp.
General Electric Co.
Hoffman Specialty Co., Inc.
Hough Shade Corp.
Illinois Engineering Co.
Independent Air Filter Co.
Johns-Manville Co.
Johnson Service Co.
Lakeside Co.
Marr-Galbreath Machinery Co.
Masonite Corp.
Moorhead-Reitmeyer Co., Inc.
Herman Nelson Corp.
J. J. Nesbitt, Inc.
New York Steam Corp.
Owens-Illinois Glass Co.
Pittsburgh Plate Glass Co.
Richmond Radiator Co.
Staynew Filter Corp.
B. F. Sturtevant Co.
Trane Co.
Union Electric Light & Power Co. of St. Louis
Waterman-Waterbury Co.
Westinghouse Electric & Manufacturing Co.
T. G. Wilson Corp.
Wood Conversion Co.
York Ice Machinery Corp.

Research During 1933

Sixteen Technical Advisory Committees were appointed by the Committee on Research to give consideration to the need for research and plans for the same in connection with as many technical subjects. These committees served both through meetings at the time of the Annual and Semi-Annual Meetings of the Society and through correspondence. Some of their reports have resulted in standardization and correlation of technical information in regard to the particular subject in which they were interested. Through their efforts programs for research were completely outlined in a number of instances and in other cases material progress was made. Be-

cause of lack of funds, active investigation of only a few of the subjects could be carried on at the Laboratory in Pittsburgh or in the co-operating universities. Prof. A. C. Willard continued his service as Technical Adviser to the Committee on Research.

Co-operation With the Universities

The Committee on Research continued the practice of enlarging its useful work through co-operation with the engineering departments of a number of universities. While it has been necessary to reduce and in many cases eliminate entirely the expenditure of funds for the support of work carried on in the universities, active co-operation has been continued with many of the institutions formerly co-operating, and co-operation has been organized in other institutions during the year. The following institutions co-operated with the Laboratory during 1933 in connection with the projects listed:

1. *University of Wisconsin:* Heating of buildings.
2. *Armour Institute of Technology:* Study of air flow through registers and grilles and air delivery by unit ventilators.
3. *University of Illinois:* Direct and indirect radiation with gravity air circulation. Also, cooling of buildings in summer.
4. *University of Kansas:* Garage ventilation.
5. *Harvard School of Public Health:* Atmospheric ionization. Also, minimum air requirements for ventilation.
6. *Yale University:* Oil burning equipment used in heating.
7. *University of Minnesota:* Dust and dust control apparatus. Also, heat transmission.
8. *Agricultural and Mechanical College of Texas:* Insulation of buildings.
9. *Case School of Applied Science:* Heat transfer from direct and extended surfaces with forced air circulation.
10. *Michigan College of Mining and Technology:* Corrosion in return lines in relation to the chemical composition of the water, vapor, and gas handled.

Papers Published During 1933

The Committee on Research through its various Technical Advisory Committees prepared 20 technical reports during 1933:

1. Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
2. Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
3. Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
4. Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw. A.S.H.V.E. TRANSACTIONS, Vol. 39. 1933.
5. The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
6. Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
7. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
8. Application of the Eupatheoscope for Measuring the Performance of Direct Radiation and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
9. Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
10. Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt, and L. C. Benjamin. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
11. Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and F. L. Broderick. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
12. Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson, and O. C. Cromer. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
13. Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933.
14. Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.
15. Comparison of Oil and Gas Firing in a Heating Boiler, by L. E. Seeley and E. J. Tavanlar. A.S.H.V.E. JOURNAL, *Heating Piping and Air Conditioning*, October 1933.

16. Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.
17. Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell, and Louis M. Farber. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.
18. 1933 Directory for the Correlation of Thermal Research. Published by the Research Laboratory-A.S.H.V.E., in booklet form under the direction of the Technical Advisory Committee on Correlating Thermal Research.
19. Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.
20. Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

Research Projects Investigated During 1933

The membership of the sixteen Technical Advisory Committees and their activities during 1933 are given below:

1. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT.—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* J. J. Aeberly, W. L. Fleisher, D. E. French, R. R. Sayers, C.-E. A. Winslow.

Activities by the committee were continued on problems relating to atmospheric and artificial ionization, and minimum air requirements for ventilation. A progress report [1] on the physiological changes during exposure to ionized air was presented at the Semi-Annual Meeting, 1933, giving the results of experiments at Harvard University in co-operation with the Laboratory. The work is being continued independently by Harvard on patients of its clinic. A second report [2] dealing with diurnal and seasonal changes in atmospheric ionization both outdoors and indoors is being presented at the Annual Meeting, 1934.

A study of minimum air requirements for ventilation from the standpoint of body odor is now being made by Harvard in co-operation with the Laboratory. This is an elaboration of the ventilation-box experiments recently made at Harvard. Two identical rooms having a small connecting door in the adjoining wall are equipped with individual unit air conditioners for maintaining the same temperature and humidity in both rooms. The control room is to be furnished with 30 or more cubic feet of air per person per minute and will be occupied by 4 to 6 trained "air smellers" who will serve as judges. The other room will be occupied by 8 to 10 subjects, receiving a varying supply of air between the limits of 5 and 30 cu ft per minute per person. After a preliminary exposure of one to two hours or more, the judges will pass back and forth from one room to the other, making comparisons between the odor intensities in the two rooms. Notes will be made of any unusual or objectionable features that may be associated with unusually low or high air flows. As a check on the foregoing readings, observations will be made of the number of dilutions necessary to reduce the odor intensity to the olfactory threshold, by means of a simple device which has been developed in connection with the ventilation-box experiments.

2. AIR FLOW THROUGH REGISTERS AND GRILLES.—*Technical Advisory Committee:* John Howatt, *Chairman;* J. J. Aeberly, L. E. Davies, D. E. French, J. J. Haines.

The work carried on by this committee was originally suggested to the Committee on Research by the *Ventilating Contractors Employers Association* of Chicago, which felt the need of a practical method of determining the rate of air delivery in an installed job. The cost of the investigation was largely borne by the contractors association, and the work was carried on by Armour Institute of Technology in co-operation with this organization and the Society. The work resulted in a paper [3] presented at the Semi-Annual Meeting, 1933. The investigation has led to the development of formulae for air flow through both supply and exhaust grilles and

[1] See No. 10 in List of Research Reports, p. 19.
[2] See No. 19 in List of Research Reports, p. 20.
[3] See No. 13 in List of Research Reports, p. 19.

a table of coefficients for use therein, which have become generally recognized and adopted as the best available practice. The publication of this year's work of the committee completes the studies originally outlined.

3. ATMOSPHERIC DUST AND AIR CLEANING DEVICES (INCLUDING DUST AND SMOKE).—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* J. J. Bloomfield, Albert Buenger, Philip Drinker, Leonard Greenburg, E. V. Hill, S. R. Lewis, H. B. Meller, Games Slayter, S. W. Wynne.

Active work was carried on by this committee during the year, both in formal meetings and through correspondence. The scope of the work included both the development of plans for laboratory work and the development of codes for testing and rating air cleaning devices for both ventilation and industrial practices. A code [4] developed by the committee for air cleaning devices used in general ventilation work was presented at the Semi-Annual Meeting 1933 and adopted with amendments for submission to the membership for approval by letter ballot.

Since the Detroit Meeting the committee has conducted investigations as to the desirability of adding a fibrous or linty constituent to the standard dust used in testing, and the possibility of more definitely defining particle sizes in the lamp black used in the standard dust. Professor Rowley, at the University of Minnesota in co-operation with the Laboratory, and S. R. Lewis, consulting engineer of Chicago, have studied the use of the test procedure for automatic air filters and have developed a performance chart. The University of Minnesota is continuing the study of the problems involved in cleaning the air of pollens and those dusts which cause allergic diseases such as hay fever and asthma.

In developing the code for air cleaning devices it became apparent that a code suitable for laboratory testing of air cleaning devices used in general ventilation work would be entirely unsuited for rating cyclone collectors, dust separators, and the like, or for the selection and rating of devices intended to supply air for dust-hazardous occupations in accordance with the standards of the U. S. Public Health Service. The use of the present code was, therefore, limited to laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air.

The committee has now under consideration a code for testing and rating air cleaning devices used in public health work. Prof. Philip Drinker has carried on preliminary work in the selection of testing dust having uniform particle sizes. The committee has received a number of requests for advice and assistance on procedures for testing air cleaners and has co-operated wherever possible.

4. CORRELATING THERMAL RESEARCH.—*Technical Advisory Committee:* R. M. Conner, *Chairman;* D. S. Boyden, J. C. Fitts, H. T. Richardson, Perry West.

The work of cataloging available literature was continued at Pittsburgh and a new Directory [5] was published in conformity with earlier plans.

5. CORROSION.—*Technical Advisory Committee:* J. H. Walker, *Chairman;* H. F. Bain, E. L. Chappell, W. H. Driscoll, R. R. Seeber.

The Technical Advisory Committee on Corrosion is advising Professor Seeber of the Michigan College of Mining and Technology regarding corrosion tests which are being undertaken under the direction of Dr. Rohrman of that institution, in co-operation with the Research Laboratory. The general plan of the work is to determine the rate of corrosion in various parts of a model heating system, operated under various conditions such as prevail in actual practice.

---

[4] See No. 7 in List of Research Reports, p. 19.
[5] See No. 18 in List of Research Reports, p. 20.

6. DIRECT AND INDIRECT RADIATION WITH GRAVITY AIR CIRCULATION.—*Technical Advisory Committee:* H. F. Hutzel, *Chairman;* A. P. Kratz, H. R. Linn, J. F. McIntire, J. P. Magos, T. A. Novotney, R. N. Trane, G. L. Tuve.

As a result of increasing interest in the application of convector heaters, the following problems have been considered by this committee for possible investigation:

(1) Is the steam condensed in a convector a true measure of its heating ability in so far as comfort temperature is concerned?
(2) If not, what tangible means of measuring heating effect can be recommended?
(3) To what extent does height of enclosure affect heating effect?
(4) Does a top outlet enclosure produce a different heating effect than a side outlet enclosure?
(5) To what extent does the volume of air circulated affect the heating effect of a convector?
(6) Is the standard correction factor formula commonly used in the case of cast-iron direct radiation adaptable to convector types of heaters when tested in the warm wall testing booths as provided for in the code for testing convector heaters?

Of these phases of the problem two have been investigated during the past year at the University of Illinois as follows:

The investigation of direct radiators and convectors has proceeded along the lines previously adopted, with tests in both the room heating testing plant and the warm wall testing booth. In connection with the room heating testing plant the construction and calibration of the eupatheoscope was completed. A study of the application of this instrument led to the recommendation of a provisional standard for its use conforming with the standards of comfort accepted in American practice. A paper [6] on the subject was presented at the Semi-Annual Meeting of the Society.

The work in the warm wall testing booth led to the conclusions that the error in the application of the correction factor specified in the A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) practically does not exceed 5 per cent when the temperature of the inlet air in the test is not more than 15 deg above or 5 deg below the 65 F adopted as a standard. These results were presented [7] at the Semi-Annual Meeting 1933.

7. GAS HEATING EQUIPMENT.—*Technical Advisory Committee:* W. E. Stark, *Chairman;* Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, H. L. Whitelaw.

Consideration was given during the year to plans for a study of heat loss through draft hoods in gas heating installations. The work was outlined and data is being collected by the Chairman of the Technical Advisory Committee in Cleveland, and by gas companies in Chicago and Washington, D. C.

8. HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION.—*Technical Advisory Committee:* F. B. Rowley, *Chairman;* H. F. Bain, H. F. Hutzel, W. G. King, A. P. Kratz, E. J. Lindseth, G. L. Tuve, W. E. Stark.

Plans for an investigation which is being carried on by Professors Tuve and McKeeman at Case School of Applied Science, Cleveland, in co-operation with the Society, were outlined by the committee and a test set-up was developed and data collected on the relation of heat emission to air circulation. A comparison of the findings with results obtained in similar studies made in England shows a greater variation with increased velocity in the case of the English studies and suggests that the minimum air velocity used in the study at Case School may not have been below the velocity of turbulence. This phase of the subject is now being investigated.

A paper for presentation to the Society is now in preparation at Case School which will present data answering the following questions:

1. If the performance of a finned-tube unit under one set of conditions is known, how can the performance under other conditions be predicted?
2. What effect has the character of the air stream upon the heat transfer?
3. Can the heat transfer of a finned-tube unit with forced air circulation be predicted by using the data already available in the publications of McAdams, Rowley, King, and others?

---

[6] See No. 8 in List of Research Reports, p. 19.
[7] See No. 11 in List of Research Reports, p. 19.

9. HEAT TRANSMISSION—(HEAT RECEIVED BY AND EMITTED BY BUILDINGS IN RELATION TO LIVING COMFORT).—*Technical Advisory Committee:* P. D. Close, *Chairman;* A. B. Algren, R. E. Backstrom, A. E. Stacey, Jr., J. H. Walker.

Plans formerly outlined for investigations of this subject at Pittsburgh and at the University of Minnesota have been largely completed. A report[8] summarizing previous work at the University of Minnesota was published. Work now being carried on in heat transmission at Minnesota includes a more critical study of the thermal conductivity of special types of wall construction and an analysis of air space conductance. An attempt is being made to make a more definite separation between the total amount of heat conducted by radiation and by convection and conduction across the air space. The surface effect on both radiation and convection is being studied and some of the problems of practical application of insulation to air spaces are included.

The committee cooperated with the Technical Advisory Committee on Refrigeration in Relation to Air Treatment in a study carried on at Pittsburgh of solar radiation through windows and the effect of curtains, awnings and Venetian blinds in its reduction, which will be reported[9] at the Annual Meeting. A paper[10] relating to the amount of total low intensity radiant energy passing through various types of glass was prepared by the Research Laboratory and will also be given at the Annual Meeting, 1934.

10. INFILTRATION IN BUILDINGS.—*Technical Advisory Committee:* D. W. Nelson, *Chairman;* V. W. Hunter, W. C. Randall, E. N. Sanbern, J. G. Shodron, Ernest Szekely.

There was a time when infiltration in buildings was regarded as a problem only during the heating season. With the widespread interest and adoption of summer cooling, the determination and control of infiltration becomes of year round importance. It is becoming increasingly necessary to know air leakage values and to improve the infiltration resistance of building units.

The proposed program of the Heat Transmission Committee points to the desirability of having a method of testing for infiltration through units such as windows in place in the rooms under investigation. This would be especially desirable in the testing of individual rooms on various floors of tall buildings. The variation in radiation required is likely largely due to a difference in infiltration. It is necessary that the windows in the various rooms selected have similar characteristics as to air leakage.

The fourteen double-hung wood windows used for the study on infiltration at the University of Wisconsin in cooperation with the Laboratory and reported on in 1931, are installed in a shelter house. They are weatherstripped and have been removed once for testing. It is proposed to test them at intervals during a five-year exposure period to determine their effectiveness under actual building exposure and heating conditions.

11. OIL BURNING DEVICES.—*Technical Advisory Committee:* H. F. Tapp, *Chairman;* Elliott Harrington, F. B. Howell, J. H. McIlvaine, L. E. Seeley.

Investigation of the plans outlined for studies of oil burners at Yale University, in cooperation with the American Oil Burner Association and the Society, was continued during the year at a reduced rate because of lack of sufficient funds. A large part of the work originally planned has been completed and results are contained in a report[11] being presented at the 1934 Annual Meeting of the Society. The program now being considered by the Committee includes the following studies:

---

[8] See No. 5 in List of Research Reports, p. 19.
[9] See No. 20 in List of Research Reports, p. 20.
[10] See No. 14 in List of Research Reports, p. 19.
[11] See No. 15 in List of Research Reports, p. 19.

1. Characteristics of a gravity hot water heating system in connection with oil burners.
2. Effect of furnace proportion on operation of various types of oil burners.
3. Effect of oil characteristics on oil burner performance.
4. Standard practice for determining fuel-burning rates of oil burners.
5. Application of burner sizes to heating load.
6. Boiler ratings for oil burner operation.

12. PIPE AND TUBING (SIZES) CARRYING LOW PRESSURE STEAM OR HOT WATER.—*Technical Advisory Committee:* S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, R. R. Seeber, W. K. Simpson.

The laboratory work outlined was largely completed and a report [12] presented to the Society. Additional work remains to be done in the correlation of the results of the laboratory studies and their resulting application to the pipe size tables of THE GUIDE.

13. REFRIGERATION IN RELATION TO AIR TREATMENT.—*Technical Advisory Committee:* A. P. Kratz, *Chairman;* E. A. Brandt, E. D. Milener, K. W. Miller, F. G. Sedgwick, J. H. Walker, R. W. Waterfill.

This project was outlined by the committee and the study carried on during the summer of 1932.[13] The work was continued during the past summer with the object of determining to what extent ice cooling in the Research Residence could be supplemented by the circulation of cool air through the house from the outdoors during the hours from 6:00 p.m. to 6:00 a.m. and also the relative merits of several methods of circulating night air.

The results of the past summer's work indicate that if the best method of supplementary cooling with night air had been consistently used during the summer of 1932 the amount of ice required for that season would have been about 10 tons instead of the 43 tons used. Even with the least favorable method of night cooling the amount could have been reduced to 23 tons. The results will be presented [14] at the 1934 Annual Meeting of the Society. The National Warm Air Heating and Air Conditioning Association, The National Association of Ice Industries, The Utilities Research Commission of Chicago, and other organizations cooperated in this study.

The committee also cooperated with the Heat Transmission Committee in a study conducted at the Research Laboratory in Pittsburgh of solar radiation through windows and the effect of curtains, awnings and Venetian blinds in its reduction, a report [15] of which is being presented at the Annual Meeting, 1934.

14. SOUND IN RELATION TO HEATING AND VENTILATION.—*Technical Advisory Committee:* Warren Ewald, *Chairman;* C. A. Andree, Carl Ashley, C. A. Booth, V. O. Knudsen, R. F. Norris, J. P. Reis, G. T. Stanton.

Consideration is being given to plans for research to standardize field methods of determining sound emanating from air conditioning equipment.

15. VENTILATION OF GARAGES AND BUS TERMINALS.—*Technical Advisory Committee:* E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore, A. H. Sluss.

Plans previously outlined by the committee were investigated during the latter part of the last heating season and this fall, resulting in two publications.[16]

---

[12] See No. 4 in List of Research Reports, p. 19.
[13] See Nos. 2, 3 in List of Research Reports, p. 19.
[14] See No. 16 in List of Research Reports, p. 20.
[15] See No. 20 in List of Research Reports, p. 20.
[16] See Nos. 9, 17 in List of Research Reports, pp. 19 and 20.

16. HEAT REQUIREMENTS OF BUILDINGS.—*Technical Advisory Committee:* D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

This committee was appointed during the year to coordinate and reconcile the many available data on heat transfer and loss, or gain, resulting from laboratory studies, with the actual formulae for using such values in the calculation of the heating requirements of a building. The work of the committee may be considered the final step in the Society's long and extended program of studying heat losses from buildings. Plans are being outlined for studies in Detroit, Pittsburgh, Minneapolis, Madison, Urbana, and other places, of the relation between calculated and actual heat requirements of buildings. At least some of these studies will include an accurate study of the hourly heat requirements to maintain desired temperature in individual rooms on different exposures and different elevations of modern buildings.

Projects now under way at the University of Wisconsin include a comparison of heating by means of direct radiation with heating by unit heaters, and studies of the economy of lowering temperatures during periods of unoccupancy and of the regulation of heat supply by zone control, by orifices, and by outside temperatures. These projects offer an excellent opportunity to obtain data useful to the Committee on Heat Requirements of Buildings concerning relations among calculated heat loss, installed radiation, actual heat loss, and the cooling-off and heating-up characteristics of building structures.

The economy of lowering temperatures during unoccupancy and the resulting peak loads on the vacuum pumps is being studied by the University of Wisconsin in cooperation with the Research Laboratory in the three-story, stone Mechanical Engineering Building which is heated by a vacuum system with direct radiation in offices and class rooms, unit heaters in most laboratories, and a central fan supply for class room ventilation. The system is equipped with a dual temperature control. During the present heating season a study will be made to determine the saving resulting from the carrying of reduced temperatures when not occupied. The maximum load occurs on the vacuum pump during the warming up periods and it is proposed to study these peak loads as related to the average load. The condensate and air removal loads will be studied separately. The discharge through calibrated orifices will be used to measure the air discharged.

The University of Wisconsin in cooperation with the Research Laboratory is making *a study of the regulation of heat supply by zone control, orifices, and by the outside temperature*. Tripp and Adams Halls are men's dormitories exactly similar except as to relation to points of the compass. A system of control of temperatures by zone valves and orifices and by radiator orifices has been installed in Adams Hall. An outside thermostat is used in determining the position of the zone valves. Condensate meters have been installed in both dormitories so that a comparison of steam consumptions may be obtained for a vacuum heating system and a similar system with the refinements of orifices and zone valves operating under control of outside temperatures.

Chadbourne Hall is a girls' dormitory heated by a one-pipe system. Condensate records have been kept over a period of years for this building. Recently a control of zone valves has been added that supplies steam intermittently in conformity to the heat losses as determined from the outside temperature. Provision is also made for reduced temperatures at night. Records are being made to determine the savings resulting from the added refinements in control.

At the conclusion of the presentation of this report, on motion of Professor Willard, seconded by Mr. Campbell, it was voted that the Report be accepted.

PRES. W. T. JONES: In closing this research program I should like to mention that I think the one activity of the Society in which more members are interested than any other is the research program. I have found this to be the case in my visits to the various Chapters and I have found very few members who object to

the fact that 40 per cent of their dues is allocated to research. Once in a while we hear a member say that he thinks the research program should be eliminated in times like these. We have received a few letters from members containing the same statement. Your officers have, of course, considered the advisability of doing away with the research program, but it seems wise not to eliminate it entirely, but to curtail it somewhat, for this reason.

At the Bureau of Mines in Pittsburgh, the United States Government assigns to us space for our Laboratory without charge. They give us heat, light, power, janitor service, in fact every service that goes with quarters in a fine building. We have access to their stores where we can buy supplies at reduced prices. We can have work done in the shops of the Bureau of Mines at low cost and the only stipulation the Government made originally for giving us this privilege was that we spend $15,000 a year on research. The contract was changed later to $10,000 a year, in view of these times.

If we abandon our research program, to which we are committed to the extent of $10,000 a year, we feel pretty confident that we would not be able to get back in the Bureau of Mines under the same favorable circumstances that exist today. It seems to your Officers a very wise move to remain there and retain that privilege just as long as we can, because, as you will note, every cent paid in by members of this Society for research goes to research. The Research Committee is operating a very extensive program with a very modest expenditure of funds.

I want to make this point for the benefit of those members who may feel that the program should be abandoned, because they will appreciate the fact that if it is abandoned now, when we put it back into effect again it will be at many times the cost of the present operating program.

Report of Committee on Code for Testing and Rating Condensation and Vacuum Pumps

The report of the Committee on Code for Testing and Rating Condensation and Vacuum Pumps [17] was made by Chairman John Howatt.

It was explained that because of the lack of unanimity of opinion or agreement about the proper conditions of rating or testing condensation and vacuum pumps, purchasers found it difficult to make a comparison of equipment from the engineering data furnished by manufacturers. Therefore, the Society in 1932 felt that a standard code was desirable and a committee was appointed with the following personnel: John Howatt, *chairman,* W. H. Driscoll, L. A. Harding and F. J. Linsenmeyer.

Desiring to prepare a code that would be universally used, the committee invited representatives of manufacturers to confer and this invitation was promptly accepted and generous support and cooperation were given. Work was started seriously in April, 1933, and the present code is one that is acceptable to the Society's committee and in general meets with the approval of the manufacturers of equipment.

Mr. Howatt then explained the objects of the code, gave the definitions which had been adopted and explained the basis of rating and the reason for selecting 160 deg water temperature and 5½ in. of vacuum.

The code was presented for adoption.

---

[17] The A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, which was adopted as amended, appears on p. 33.

The following amendments were presented and approved. C. V. Haynes proposed an amendment to the title, so that it would read:

American Society Heating and Ventilating Engineers Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps.

He also suggested a change in paragraph 2 of the Definitions, so that it would read:

A Vacuum Pump for the purpose of this Code shall be construed to mean a return line vacuum pump for steam heating systems with a return vacuum up to 10 in. of mercury.

Howell Adams suggested a change in paragraph 8, Section 3, so that it would read as follows:

Horsepower input into the pump during the period when tests (1) and (2) are being taken. Note: If during the test the pump is motor-driven, the motor driving the pump shall be rated in accordance with standards established by the *A. I. E. E.* with allowed corrections for conventional efficiency.

A motion by F. D. Mensing, seconded by Mr. Haynes, provided that paragraph 8, section (1), be amended to require a test run of one hour. After considerable discussion on this point by Messrs. Ashley, Wiley, Howatt and Mensing, on vote the motion was lost.

A vote was then taken on the motion to adopt the code as amended, which was made by C. F. Ames and seconded by Mr. Haynes. Upon passage of this motion, President Jones announced that the code would be submitted to the entire society membership for vote by letter ballot in the manner prescribed by the Constitution and By-Laws.

On motion of Mr. Haynes, seconded by Russell Donnelly, it was voted that the Code Committee be continued and that it be instructed to prepare a test code for return line vacuum pumps in excess of 10 in. of mercury.

At the third session, Tuesday, February 6, 2:00 p. m., President Jones introduced W. L. Fleisher, chairman of the Guide Publication Committee, who presented the following report.

Report of Guide Publication Committee

THE GUIDE 1934 is about ready for distribution to the membership and copies will be mailed within the next 10 days. This year THE GUIDE again appears in a flexible red binding and contains 42 chapters of technical data covering 618 pages with all subjects cross-indexed. One hundred manufacturers used catalog data space totaling 176 pages with an Index to Modern Equipment. The first printing was limited to 7500 copies and the income from advertising totals $15,300, and the estimated return from copy sales will be $8500. The cost of THE GUIDE 1934 for printing, distributing and all other services was about $18,500 to date.

The Technical Data Section of this 12th Edition has been enlarged to include newly developed data that are vitally important in meeting the present day demands of engineers who devote their time to heating, ventilating and air conditioning practice. From the practical experience of members as well as from available research sources useful facts have been gathered and incorporated in the 42 chapters which have been arranged for convenient reference. An extensive index has been included to aid the reader. The text is prepared for engineers, architects, contractors and students

who are designing, operating, specifying, installing and studying systems and apparatus the functions of which are to create comfort and to improve the efficiency of processing.

For the important work of creating THE GUIDE 1934 the assistance of engineers of specialized training and knowledge was enlisted. The careful reviewing, compiling and editing of data selected from authoritative sources was accomplished through the cooperative efforts of the following men who have unselfishly given of their time and knowledge:

T. Napier Adlam
J. C. Albright
H. L. Alt
C. F. Ames
C. M. Ashley
A. L. Baum
J. L. Blackshaw
C. A. Bulkeley
R. E. Cherne
L. A. Cherry
P. D. Close
R. F. Connell
J. M. DallaValle
Philip Drinker
M. William Ehrlich
John Everetts, Jr.
F. H. Faust
W. G. Frank
R. S. Franklin
F. E. Giesecke
W. C. Goodwin
L. A. Harding
H. M. Hart
F. C. Houghten
L. P. Hynes
R. E. Keyes
V. O. Knudsen
A. P. Kratz
J. W. Kreitner
G. L. Larson
R. D. Madison
J. F. McIntire
H. B. Meller
J. G. Mingle
H. C. Murphy
D. W. Nelson
A. J. Nesbitt
M. E. O'Connell
A. J. Offner
E. C. Rack
F. G. Sedgwick
G. L. Tuve
C. P. Yaglou
John Zink

The Guide Publication Committee is glad to acknowledge the splendid cooperation of these engineers who have served so willingly for the benefit of their associates in the profession. The Society, too, should thank these men for the assistance that they gave the Committee and for the time that they spent in the service of the organization.

Some specific suggestions have been made to the council regarding future handling of THE GUIDE in order that its potentialities may be realized. One particular point however, that might be stressed here is that THE GUIDE provides the most extensive contact with individuals and groups outside of the organization and it has spread the name of the A. S. H. V. E. throughout the world. THE GUIDE has been instrumental also in carrying the results of our Research Laboratory work to thousands outside of our Society membership and it has aided materially in giving up-to-date data on air conditioning which today is about the third most talked of subject in engineering, the others being automobiles and airplanes.

It is the opinion of your Committee that THE GUIDE has unlimited future possibilities and it should be arranged and edited to serve the large group of people who are conscious of and interested in air conditioning. It is hoped that some way funds can be found to do this.

In conclusion I wish to thank the members of the Committee for the support that they have given me in handling the work which was a far bigger job than I had any conception of and to express the hope that when the members see THE GUIDE 1934 that they will find pleasure in using this new volume.

Respectfully submitted,

GUIDE PUBLICATION COMMITTEE,

W. L. FLEISHER, *Chairman.*

The fourth session of the Society's Annual Meeting was called to order on Wednesday, February 7, 9:30 a. m., by President Jones, who introduced the chairman of the Membership Committee, E. K. Campbell, of Kansas City. Mr. Campbell presented his report.

Report of Membership Committee

While there are some encouraging factors in the membership situation your Committee feels there are still many things to be accomplished during the coming year.

President Jones was very helpful in his visits to the Chapters and he made definite progress in reselling this organization to its membership. This has been badly needed and some more work along this line must be done. When members are fully sold on their Society, the problem of membership will take care of itself.

It is a true statement that there are thousands of men over the country waiting for an invitation to join the Society and this is manifest by the many who write into the headquarters office inquiring about the requirements for membership. These men are listed and if the prospect is in Chapter area, notice is sent to the local membership chairman. Where prospects are outside of chapter areas, the Membership Committee follows up the matter.

As the work of the Society becomes better known there are many who feel the urge to join it, and the work of finding eligible men must be done by the rank and file of the membership, as it is impossible for a small Membership Committee, even with the fine cooperation that is given by local chapter membership committees to contact all of the people who would make desirable members.

During 1933, 256 applications were received, of which 84 were from students; 102 resigned and 300 were dropped. There was a net loss of 230 in membership and a corresponding loss of income to the Society.

Student membership in the opinion of your Committee is a good investment and a future asset, but is a present liability. Of the number dropped during the year nearly half should have been canceled in the previous year, but the Council was loath to drop them from the rolls until all efforts to keep them had been exhausted.

It can be seen that this year with only 172 candidates elected to Member, Associate and Junior grades, it is insufficient to keep the Society going and the dues collected from our members now on the rolls, consisting of 1268 Members, 363 Associates, 131 Juniors and 125 students has not paid for the services that they are receiving. Research activities have been particularly hard hit and the amount expended has been reduced about 60% in the past two years. Because of the publications, the Society has continued to maintain much of its usual service to the membership.

As Chairman of the Membership Committee, it is my observation that the average member is indifferent in his national outlook and at this critical time there should be a strenuous effort made to not only resell our membership on the Society but also use some salesmanship in interesting eligible candidates.

Respectfully yours,

MEMBERSHIP COMMITTEE,

E. K. CAMPBELL, *Chairman.*

With President Jones in the chair, Robert D. Williams, president of the *Heating, Piping and Air Conditioning Contractors National Association,* was introduced and extended greetings and best wishes for a successful meeting.

President Jones opened the fifth session on Wednesday, February 7, 2:00 p. m., and said the first presentation of the afternoon would be the Report of the Publication Committee by W. E. Stark.

Report of Publication Committee

The Publication Committee is composed of three very diligent readers. We are surpassed in our diligence as readers only by the chairman of the Guide Publication Committee, who reads everything that goes into THE GUIDE, and by Mr. Hutchinson, the Secretary, who reads everything that goes into THE GUIDE and every publication as well.

One of the nominal duties of the Publication Committee is the preparation and publication of THE TRANSACTIONS. However, the Publication Committee's work in connection with THE TRANSACTIONS is often more nominal than real, because our

very efficient Secretary does all of the work connected with THE TRANSACTIONS. Two volumes, that for the year 1931 and that for the year 1932, are nearly completed and will be sent to members in about a month.

One of the members of the Publication Committee presented a paper this morning, the paper on the Thermodynamics of Air Conditioning, and he stated before he started presentation that he didn't know very much about the thermodynamics of air conditioning, but he was really fooling you a little bit because he got a chance to learn something of the subject when he went over that paper.

The report of the Publication Committee may be summarized for purposes of record. During the year the Committee reviewed a total of 35 papers with the idea of upholding the quality of presentations recorded in THE TRANSACTIONS of the Society and providing technical information of interest and value in useful and concise form. Ten papers were presented at the 1933 Semi-Annual Meeting in Detroit and 20 papers have been scheduled for this New York Annual Meeting of 1934. A few papers were rejected as being unsuitable for Society publication and two were withdrawn.

President Jones said that he was glad to announce a registration of over 500 at the meeting with members and guests from Canada, Oklahoma, Florida and Texas and expressed the belief that Professor Giesecke held the long distance record from College Station, Texas.

President Jones said that one of our members had not registered this year for the first time since the Society was organized in 1895. He attended the 39th Annual Meeting in Cincinnati, left the sessions because of illness, returned home, and in a month his name was removed from our roll of membership and transferred to the celestial society above. This wonderful record of having attended every Annual Meeting of the Society since its inception in 1895 belonged to Andy Edgar of Philadelphia.

The final session of the 40th Annual Meeting was called to order on Thursday, February 8, 2:00 p. m., by President Jones who announced that the newly elected officers should be installed.

At the request of President Jones, Thornton Lewis escorted Roswell Farnham one of the retiring Council members to the rostrum. President Jones on behalf of the Society expressed his thanks for the seven years service which had been rendered by Mr. Farnham.

W. H. Driscoll conducted C. V. Haynes, President-elect, to the rostrum and President Jones said that as executive head of the Society for the coming year a great measure of responsibility rests upon your shoulders and because of your familiarity with the organization and the duties to be performed as president and your enthusiasm for the Society, we feel proud to welcome you as President.

First Vice-President John Howatt was conducted to the rostrum by W. H. Driscoll and he was installed by Thornton Lewis.

The Second Vice-President, Prof. G. L. Larson, was escorted to the rostrum by Thornton Lewis for installation by Mr. Driscoll.

The newly elected Council members, who were present, Messrs. Beman, Gurney and Russell, were brought to the rostrum by W. H. Carrier and were installed by Mr. Jones.

The gavel was turned over to President-elect C. V. Haynes, who expressed his pleasure for the honor conferred upon him and then inquired whether any new business was to come before the meeting.

W. R. Eichberg presented a petition signed by the required number of members, proposing an amendment to Article B-VIII, Section 1, of the By-Laws. W. H. Driscoll pointed out that the proposed amendments should take the regular procedure, so that an opportunity might be given to all members to vote for or against it.

Mr. Jones suggested that the resolution be introduced and submitted to the Council for reference to the Committee on Constitution and By-Laws. On motion of Mr. Jones, properly seconded, it was voted unanimously to refer the resolution to the Council.

E. H. Gurney asked the privilege of the floor and offered a motion extending the sincere thanks of the Society to Past President Jones for his splendid service during the past year. The motion was seconded and carried by a rising vote.

Report of Committee on Nomenclature

W. H. Driscoll was introduced and read the report of Samuel R. Lewis, chairman of the Committe on Nomenclature. A letter from the committee to Mr. Jones was read and Mr. Driscoll pointed out that the Committee on Nomenclature was appointed immediately after the Annual Meeting in Cincinnati to eliminate from our literature the term, *square foot,* and substitute something that would be more accurate and more descriptive. He explained about the animated discussion and correspondence that took place between members of the committee and reviewed the conference of organizations held in Detroit under the auspices of the Society.

At this conference a resolution was adopted to the effect that it was the opinion of those present that the symbols, Mb and Mbh, be adopted and it was agreed that the representatives of the various organizations would go back to their respective societies and endeavor to have them adopt the proposed symbols. On the following day, the Society in session at Detroit approved the symbols proposed by the Committee on Nomenclature, Mb to express 1000 Btu and Mbh to express 1000 Btu per hour.

Subsequently correspondence came from the *A. S. M. E.* indicating that its representatives at the Detroit meeting were not familiar with some of the previous activities, as an *A. S. M. E.* committee had recommended the use of *kB* to express 1000 Btu. After some correspondence, the committee has decided that it might be advisable to adopt the *A. S. M. E.* symbol, but before that can be done, it is necessary to rescind the action taken at the Detroit meeting. If this is done, the next step would be the adoption of a new symbol, so in order to get the matter before the meeting, the following resolution was offered:

Resolved, that the Society rescind the action taken at the Semi-Annual Meeting, 1933, in Detroit with respect to the adoption of the symbols Mb and Mbh.

The motion was seconded by Mr. Jones.

A motion to lay the matter on the table, pending the result of the committee's conference with interested organizations, was offered by W. L. Fleisher, seconded by J. D. Cassell, and carried by vote to those present.

Report of Committee on Revision of Constitution and By-Laws

President Haynes called upon the Secretary to present the Report of the Committee on Revision of Constitution and By-Laws.

Secretary A. V. Hutchinson reported that recommendations of the Committee on Constitution and By-Laws, approved by the Council, had been sent to the membership in accordance with the procedure required by Article B-XVI of the By-Laws. The changes are as follows:

Art. B-IV Sect. 4—Of the annual dues paid by the members of each grade, three dollars ($3.00) shall be considered as a subscription for the Journal of the Society.

To be amended to read—Art. B-IV Sect. 4—Of the annual dues paid by members of each grade, *a sum equal to its current subscription price* shall be considered as a subscription for the Journal of the Society.

Art. B-XI Sect. 6—After December thirty-first, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council at its last meeting in the calendar year. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee, and published in the Society Journal.

To be amended to read—Art. B-XI Sect. 6—After December thirty-first, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee, and published in the Society Journal.

On motion of Professor Larson, seconded by Mr. Cassell, the vote was unanimous in favor of the changes.

Resolutions

President Haynes called upon W. E. Stark, Chairman of the Resolutions Committee, for his report which was as follows:

Whereas, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in Annual Meeting assembled in the City of New York for the period of February 5-9, 1934, inclusive, is about to bring its sessions to a close, and

Whereas, it is altogether fitting and proper as well as a distinct pleasure to remember those agencies, which have so materially contributed to the benefit and pleasure derived from that meeting, therefore

Be It Resolved, that we extend to the press of the City of New York and to the technical and trade press, our appreciation of the publicity extended to the Society in bringing its work to the attention of the public, and

Be It Further Resolved, that we accord to Mr. Paul Whiteman an expression of our appreciation of his personal appearance with his orchestra on the occasion of our President's Reception, and

Be It Further Resolved, that we wish to convey to the management of the Hotel Biltmore our thanks for the courteous attention rendered, particularly in view of the many difficulties encountered at the time, and

Be It Further Resolved, that we take pleasure in congratulating Mr. Charles Roth and his associates on the success attending their management of the Third International Heating and Ventilating Exposition, which was held coincidentally with our Annual Meeting, and

Be It Further Resolved, that we assure Mr. E. P. H. James, the principal speaker at our Annual Banquet, of our appreciation of his entertaining discourse on that occasion and that we convey to the management of Radio City our thanks for the

privilege extended our members and guests to inspect the many interesting mechanical and cultural features of that institution, and

Be It Further Resolved, that copies of these resolutions be sent to the New York Chapter, to the newspapers and trade press, to the management of Radio City, to Mr. Paul Whiteman, to the management of the Hotel Biltmore, to Mr. Charles Roth and to Mr. E. P. H. James.

The motion to adopt the resolutions offered was made, seconded and unanimously carried.

There being no further business, the motion to adjourn was offered by D. S. Boyden, seconded by Thornton Lewis, and carried.

PROGRAM 40TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL BILTMORE, NEW YORK, N. Y.
FEBRUARY 5-9, 1934

Monday, February 5

9:00 A.M. Registration (Fountain Room, 19th Floor, Hotel Biltmore)
9:30 A.M. Meeting of Tellers of Election (Room 1938)
10:00 A.M. Meeting of the Council (Room 1940)

1:00 P.M. Opening of Third International Heating and Ventilating Exposition at Grand Central Palace (46th St. and Lexington Ave.)
2:00 P.M. Pump Manufacturers Engineering Committee (Room 1938)

2:00 P.M. FIRST SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)
Greeting
Response by President W. T. Jones

Technical Papers:

Automatic Controls for Forced-Air Heating Systems, S. Konzo and A. F. Hubbard
Heating Buildings with Hot Water, B. F. Burt and S. R. Lewis

Report of the Council
Report of the Secretary
Amendments to By-Laws
Report of Finance Committee, John Howatt, *Chairman*
Report of Tellers of Election, E. E. Ashley, *Chairman*

1:00 P.M. to 10:30 P.M. Heating and Ventilating Exposition, Grand Central Palace
7:00 P.M. Past Presidents' Dinner, Transportation Club, 18th Floor, Hotel Biltmore
10:00 P.M. President's Reception and Get-Together Supper and Informal Dance, Paul Whiteman and his Orchestra, Casino Bleu, Lobby Floor, Hotel Biltmore

Tuesday, February 6

9:30 A.M. SECOND SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)
Report of Committee on Research, Prof. G. L. Larson, *Chairman*

Technical Papers:

Design and Equipment of the Pierce Laboratory, C.-E. A. Winslow, Leonard Greenburg, M.D., L. P. Herrington and H. G. Ullman
Selecting Temperatures and Wind Velocities for Calculating Heat Losses, P. D. Close

Radiation of Energy Through Glass, J. L. Blackshaw and F. C. Houghten
Studies of Solar Radiation Through Bare and Shaded Windows, F. C. Houghten, Carl Gutberlet and J. L. Blackshaw
Report of Committee on Code for Testing and Rating Condensation and Vacuum Pumps, John Howatt, *Chairman*

12:00 NOON to 10:30 P.M. Heating and Ventilating Exposition, Grand Central Palace

2:00 P.M. THIRD SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)
Report of Guide Publication Committee, W. L. Fleisher, *Chairman*

Technical Papers:

A Proving Home for Air Conditioning Investigations, Elliott Harrington and Leon A. Mears
Low-Cost Air Conditioning for a Small Residence, M. K. Drewry
Comfort Cooling with Attic Ventilating Fans, G. B. Helmrich and G. H. Tuttle
Study of Summer Cooling in the Research Residence for the Summer of 1933, A. P. Kratz and S. Konzo

12:00 NOON to 10:30 P.M. Heating and Ventilating Exposition, Grand Central Palace

2:30 P.M. Ladies' Bridge and Tea (Room 1939, Hotel Biltmore)

7:30 P.M. Meeting of Committee on Research (Room 1940)

Wednesday, February 7

9:30 A.M. FOURTH SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)
Report of Membership Committee, E. K. Campbell, *Chairman*

Technical Papers:

Thermodynamic Properties of Moist Air, J. A. Goff
Study of Air Conditioning and Office Employees Efficiency, W. J. McConnell, M.D., and I. B. Kagey
Observations of Hay-Fever Sufferers in Air-Conditioned Room and the Relationship Between the Pollen Content of Outdoor Air and Weather Conditions, T. A. Kendall and Garland Weidner, M.D.
Corrosion Studies in Steam Heating Systems, R. R. Seeber, F. A. Rohrman and G. E. Smedberg

Report of Publication Committee, W. E. Stark, *Chairman*

12:00 NOON to 10:30 P.M. Heating and Ventilating Exposition, Grand Central Palace

2:00 P.M. FIFTH SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)

Technical Papers:

Carbon Monoxide Surveys of Two Garages, A. H. Sluss, E. K. Campbell and L. M. Farber
Diurnal and Seasonal Variations in the Small-Ion Content of Outdoor and Indoor Air, C. P. Yaglou and L. C. Benjamin
Air Conditioning in Its Relation to Human Welfare, C. A. Mills, M.D.
Engineering in the Hospital of Tomorrow, C. F. Neergaard

7:30 P.M. Annual Banquet and Dance, Meyer Davis Music, Cascade Ballroom, 19th Floor, Hotel Biltmore

Thursday, February 8

9:30 A.M. Assemble at Radio City, Information Desk, Fifth Ave. Plaza Entrance, for Inspection of Mechanical Equipment in N.B.C. and British Buildings

11:00 A.M. Meeting of the Nominating Committee (Room 1940)

12:00 NOON to 10:30 P.M. Heating and Ventilating Exposition, Grand Central Palace

1:30 P.M. Ladies' Theatre Party at Radio City Music Hall. (Assemble at Vanderbilt Ave. entrance, Ground Floor, Hotel Biltmore)

2:00 P.M. SIXTH SESSION (Cascade Ballroom, 19th Floor, Hotel Biltmore)

Technical Papers:

Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, L. E. Seeley and E. J. Tavanlar

Design and Valuation of Cast Iron Domestic Heating Boilers, C. W. Brabbée

Conference on Nomenclature

Installation of Officers

Resolutions

Adjournment

4:00 P.M. Organization Meeting of the Council (Room 1940)

Friday, February 9

9:30 A.M. Meeting of Manufacturers of Heating, Ventilating and Air Conditioning Industry (Cascade Ballroom, 19th Floor, Hotel Biltmore)

Discussion of NRA Codes. Speakers: William H. Davies, National Compliance Director, NRA, and Beverly S. King, Assistant Deputy Administrator

COMMITTEE ON ARRANGEMENTS

ARTHUR RITTER, *General Chairman*

Finance: F. E. W. Beebe, *Chairman;* J. B. Hashagen, W. M. Heebner.

Banquet: H. B. Hedges, *Chairman;* Russell Donnelly, Walter Heibel, A. J. Offner, E. J. Ritchie.

Reception: W. W. Timmis, *Chairman;* E. A. Bennett, F. B. Campbell, M. E. Durkee, H. B. Eells, Margaret Ingels, H. G. Meinke, G. E. Olsen, W. F. Rainger, Louis Hament, H. T. Richardson, E. J. Ritchie, E. S. White.

Entertainment: Alfred Engle, *Chairman;* F. B. Campbell, H. W. Fiedler, C. H. Quirk.

Ladies: C. S. Hoffman, *Chairman;* C. R. Place, Mrs. C. S. Hoffman, Mrs. C. R. Place.

Radio City Inspection Trip: W. L. Keplinger, *Chairman;* C. S. Hoffman, *Vice-Chairman.*

Publicity: R. B. Purdy, *Chairman;* Clifford Strock.

No. 969

A. S. H. V. E. STANDARD CODE FOR TESTING AND RATING RETURN LINE LOW VACUUM HEATING PUMPS

COMMITTEE: John Howatt, *Chairman;* W. H. Driscoll, L. A. Harding, F. J. Linsenmeyer

A. OBJECT OF THE CODE

1. The object of the Code is to provide a standard method of testing and rating the air and water capacity of vacuum heating pumps under standard conditions as hereinafter defined.

B. DEFINITIONS

2. A Vacuum Pump for the purpose of this Code shall be construed to mean a return line vacuum pump for steam heating systems with a return vacuum up to 10 in. of mercury.

3. The term Vacuum for the purpose of this Code is construed to mean the difference between the absolute pressure maintained at the pump inlet connection from the heating system and the absolute pressure of the atmosphere, measured in inches of mercury, referred to standard barometer.

4. Pump Head Pressure is construed as the total pressure head against which the water from the pump is discharged measured in pounds gage pressure in pounds per square inch at the discharge outlet of the pump.

5. Entering temperature is the temperature of the gases and temperature of the water as hereinafter defined measured at the pump inlet connection from the heating system and expressed in degrees Fahrenheit.

6. Standard Air for the purpose of this Code is air weighing 0.0750 lb per cubic foot. This weight corresponds to the weight of dry air at a temperature of plus 70 F and an absolute pressure of 29.92 in. mercury.

7. Horsepower is construed as the brake horsepower supplied to the pump shaft.

C. BASIS OF RATING

8. Rating Factors to be specified. The rating of a vacuum pump shall specify the following:

(1) Maximum capacity of water at 160 F in gallons per minute as determined by not less than a 20 min run after conditions have become stabilized.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., Feb., 1934, and adopted as amended.

(2) Simultaneous capacity of the pump to handle saturated air and water at 160 F and at 5½ in. of mercury vacuum, the saturated air expressed in terms of cubic feet per minute at said temperature and vacuum and the water in gallons per minute as determined by not less than a 20 min run after conditions have become stabilized.

(3) Horsepower input into the pump during the period when tests (1) and (2) are being taken. (*Note:* If during the tests the pump is motor driven the motor driving the pump shall be rated in accordance with standards established by the *A. I. E. E.* with allowed corrections for conventional efficiency.)

9. The Standard Basis of Rating is to be the rating for the simultaneous air and water capacity of the unit per minute and the power input when operating under a condition of a vacuum of 5½ in. of mercury at the pump inlet connection to the heating system handling condensate and saturated air at a temperature of 160 F and discharging the condensate against a fixed stated head of 20 lb gage.

10. Rating tables for Vacuum Pumps shall contain in tabular form all of the data specified in Par. 8 of this Code.

11. Tolerance. As there are errors in reading instruments of the type used in these tests, a tolerance of plus or minus 2½ per cent will be permitted in test results.

D. METHOD OF TEST

12. The air and vapor capacity of the pump shall be measured by a flange type velocity head flow meter employing a concentric thin plate square edged orifice of non-corrosive material as recommended in the report of the *A. S. M. E.* Special Research Committee, part 2, on Fluid Meters published and accepted by the *A. S. M. E.* in 1931. The meter shall be placed in the test apparatus so it will measure the free air from the room entering the saturating tank. Only approved calibrated meters of approved manufacture shall be used. The 5½ in. of mercury vacuum required to be maintained at the pump inlet shall be controlled by a regulating valve placed between the air meter and the saturating tank.

13. The water capacity of the pump shall be measured by calibrated meters of known accuracy compensated for hot water flow, or calibrated tanks.

14. The partial vacuum under which the pump is operating shall be measured by means of a mercury manometer with one leg of the manometer connected to the suction line on the inlet side of the pump strainer and the other leg of the manometer open to the atmosphere.

15. The drop in pressure across the air measuring orifice shall be measured by a water manometer with the legs connected at opposite sides of the measuring orifice placed in accordance with the requirements of the particular orifice meter used.

16. The temperature of the air and water supply shall separately be obtained by thermometers reading to 1 F inserted in the water line and saturated air line near their junction near the pump inlet.

17. The degree of saturation of the air shall be determined by calibrated wet and dry bulb thermometers reading to 1 F inserted in the saturated air line near the junction of the air and water lines. A suitable water well with wick to mercury bulb shall be provided for the wet bulb thermometer. The relative humidity of the free air shall be determined by wet and dry bulb thermometers.

18. The pressure head against which the pump is to discharge water shall be maintained at a fixed amount by a valve adjusted to maintain 20 or 30 lb as desired. A calibrated pressure gage connected into the pump discharge line near the pump shall be used to determine the discharge pressure.

19. The desired vacuum under which the pump shall operate shall be obtained by a regulating valve on the saturating tank.

E. TEST PROCEDURE

20. Select a measuring orifice of such a size that the drop in pressure through the orifice shall be between 2 in. and 6 in. of water, and install this orifice in the air measuring unit.

21. Select a water meter or calibrated tanks of the proper size for the

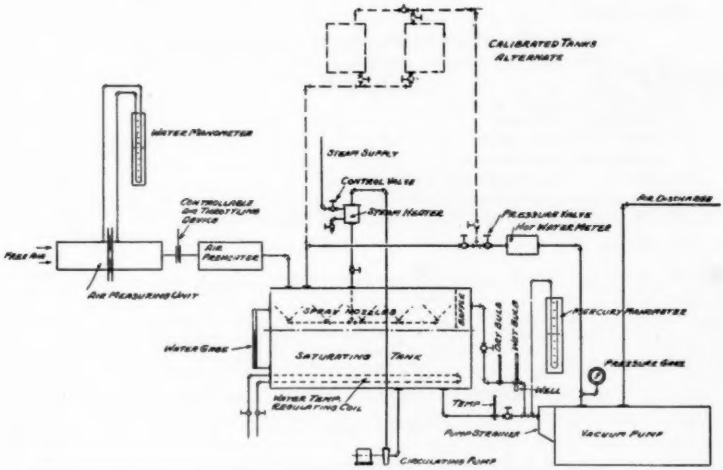


FIG. 1. TEST SET-UP FOR LOW VACUUM PUMPS

pump to be tested within the range of calibration for accuracy and install in pump discharge line.

22. Connect up and put in place all measuring instruments as indicated on the diagrammatic sketch of the approved test arrangement, Fig. 1, including calibrated electrical instruments or dynamometer to test the power input to the pump.

23. Adjust the vacuum at the pump, the temperature and flow of the water and air and the pressure head to the values desired. Operate the pump unit for 10 min under the fixed condition of the test before commencing to take test readings. Maintain conditions constant within plus or minus 1 per cent of the desired values throughout the test run. Read and record all instruments every

5 min for a period of not less than 20 min. At the close of the test take the average of the readings for the readings of record.

SAMPLE TEST CALCULATION

Assume average readings on a test run as follows:

Room barometric pressure	29.00 in. mercury
Room air temperature	70 F
Relative humidity of air in rooms	50 per cent
Pump vacuum at inlet	5½ in. mercury
Entering temperature of saturated air to pump	160 F
Orifice differential pressure	2.5 in. of water
Orifice diameter	0.52 in.
Orifice constant	0.602
Pump head gage pressure	20 lb
Coefficient of the orifice	0.602

To be determined:

The quantity of air handled by the pump measured in cubic feet per minute at 160 F and 5½ in. vacuum.

FORMULA $$W = 60\,C\,A\sqrt{2g\,d\,(p_1 - p_2)}$$

W = pound per minute flowing
C = discharge coefficient
A = area of orifice square feet
g = acceleration due to gravity
d = density of air at entering conditions
p_1 = abs. pressure entering orifice pounds per square foot
p_2 = abs. pressure leaving orifice pounds per square foot

To find density of air:

Density of vapor $\frac{0.50}{869} = 0.000576$ lb. per cubic foot

Vapor pressure $= 0.7386 \times 0.5 = 0.369$ in. mercury

Air pressure $= 29.000 - 0.369 = 28.631$ in. mercury

$$\frac{1}{V} = \frac{p}{RT} = \frac{28.631 \times 0.491 \times 144}{53.3 \times 530} = 0.0717 \text{ lb cubic foot}$$

$0.0717 + 0.000576 = 0.072276$ lb cubic foot mixture

$$A = \frac{\pi}{4}\,\frac{(0.52)^2}{144} = 0.00148 \text{ square foot}$$

$$p_1 - p_2 = \frac{2.5 \times 144}{2.307 \times 12} = 13.0 \text{ lb square foot}$$

$$W = 60 \times 0.602 \times 0.00148\sqrt{64.34 \times 0.072276 \times 13}$$
$$= 0.415 \text{ lb per minute}$$

cfm = 5.74 entering orifice

Convert to 160 F and 5.5 in. vacuum

$29.00 - 5.5 = 23.5$ in. absolute entering orifice

9.649 in. mercury vapor pressure at 160 F

$23.5 - 9.649 = 13.851$ in. partial air pressure entering pump

28.631 in. partial air pressure entering orifice

Hence $5.74 \times \frac{620}{530} \times \frac{28.631}{13.851} = 13.90$ cfm entering pump in saturated condition

No. 970

AUTOMATIC CONTROLS FOR FORCED-AIR HEATING SYSTEMS

By S. KONZO* *(MEMBER)* AND A. F. HUBBARD† *(NON-MEMBER)*, URBANA, ILL.

The results presented in this paper were obtained in connection with an investigation of coal-fired warm air furnaces and heating systems conducted in cooperation with the National Warm Air Heating and Air Conditioning Association by the Engineering Experiment Station of the University of Illinois, of which Acting Dean A. C. Willard is the Director. This work was carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation and head of the department, and A. P. Kratz, Research Professor. The results will ultimately be incorporated in a bulletin of the Engineering Experiment Station.

THE principal objects of this study were (1) to determine what particular features are essential to a good control system and (2) to compare the performance characteristics of several different types of control systems and to classify the systems from the viewpoint of satisfactory performance. In all, eleven different control systems have been tested in connection with the coal fired, forced air heating plant in the Research Residence and this study has indicated certain desirable features which any automatic control system must possess in order to insure satisfactory operation of the plant as a whole. From this standpoint, a good control must:

1. Maintain constant and uniform house temperatures for all weather conditions.
2. Be dependable.
3. Be simple in construction and operation.
4. Maintain a uniform combustion rate just sufficient to balance the heating demands on the plant.
5. Handle sudden demands on the heating plant without allowing the furnace temperatures to increase beyond a safe limit.
6. Entail a minimum of adjustment by the home owner.
7. Provide for safety features to check combustion in the event of electrical or mechanical failures.

DESCRIPTION OF PLANT

All of these tests were run under actual operating conditions on the forced air heating plant in the Research Residence during the winters of 1932 and

* Special Research Associate in Mechanical Engineering, University of Illinois.
† Research Graduate Assistant in Mechanical Engineering, University of Illinois.
Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by A. F. Hubbard.

1933. The Residence is a three-story structure of standard frame construction and is not insulated or equipped with either storm sash or weather stripping. The total space heated consisted of four rooms and a hallway on each of the first two stories and two rooms and a hallway on the third story, making in

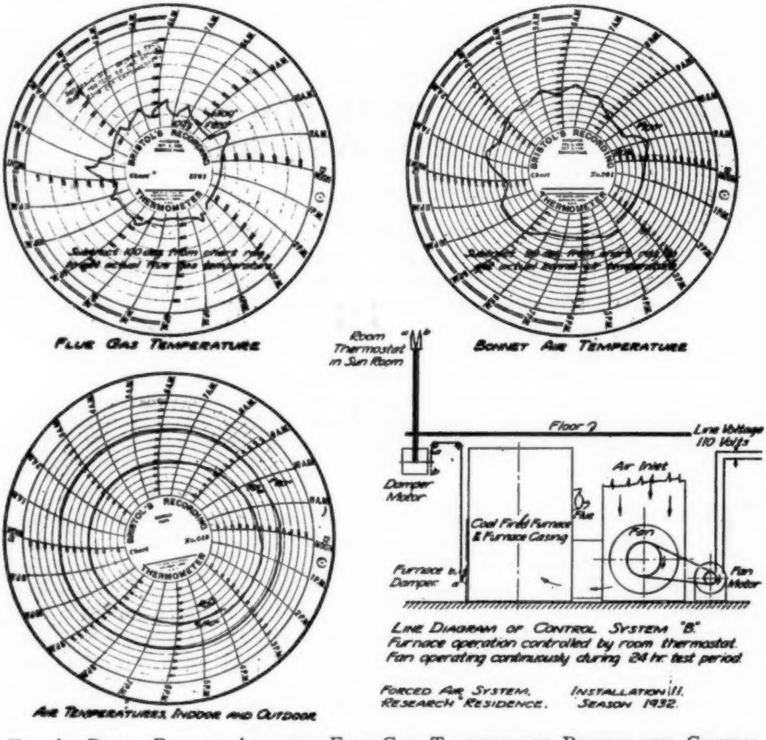


FIG. 1. ROOM, BONNET AIR, AND FLUE-GAS TEMPERATURE RECORDS FOR CONTROL TYPE I

all 17,540 cu ft. The heat loss with an indoor-outdoor temperature difference of 80 F was approximately 159,000 Btu per hour, exclusive of the basement loss.

The heating plant consisted of a coal-burning, cast-iron, circular radiator furnace; a duct system consisting of flat rectangular leaders running along the basement ceiling to conventional risers and baseboard registers used in connection with the former gravity installation; and a centrifugal fan, located in the cold air return duct and used for the purpose of forcing the air through the heating plant. The amount of air delivered by the fan during these tests was approximately 1600 cfm, exclusive of a few tests which were run with approximately 2000 cfm. Integrating and indicating wattmeters were placed in the fan motor circuit in order to record the power consumption, and a

self-starting electric clock was used to record the total length of time that the fan was actually running during a 24-hour test period. The fan motor used was of the repulsion-induction type.

The fuel burned in these tests was stove-sized anthracite coal, which had a heating value of 13,590 Btu per pound as fired. A few tests were also conducted in which an Illinois bituminous coal of high volatile content was used. Since the operating characteristics when using solid fuels are decidedly different from those obtained when using liquid or gaseous fuels fired intermittently, this paper covers specifically only hand-fired, coal-burning furnace plants.

All of these tests were run on the same heating plant; hence the results were dependent only on the changes made in the control system. In every

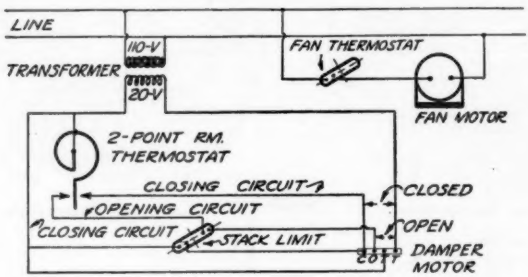


FIG. 2. WIRING DIAGRAM FOR CONTROL TYPE I. (STACK LIMIT THERMOSTAT INCLUDED)

case, the room thermostat was located in a first story room, at breathing level (5 ft) height.

In the design of the duct system for a forced-air heating plant, the assumption is tacitly made that the fan operation will be continuous and that a constant quantity of air under pressure will be delivered through the duct system. The duct system is then designed so that the short runs have greater frictional resistance to air flow per 100 ft of duct than longer runs in order to equalize the total frictional resistance for all runs. This procedure is directly opposite to that employed in the design and operation of a duct system for all-gravity operation, where the size of first story leaders is larger than that of second and third story leaders for equivalent heat carrying capacities. Since a large number of forced-air plants are operated with the fan running intermittently, with part-gravity flow and with part-fan operation, rather than with the fan running continuously, the dampers in the leaders must necessarily be adjusted to a position that is a compromise between that for all-gravity and that for all-fan operation. If the dampers are set for this medium position, they may not be in proper adjustment for all weather conditions, and at the extreme conditions of operation may allow unequal temperatures to exist among the different rooms of the house. The use of certain types of controls for intermittent fan operation aggravated this condition, as will be pointed out later.

PERFORMANCE CHARACTERISTICS

With the exception of one or two minor variations, the operation of each of the eleven different control installations, which were designated as *A, B, C,* etc., was characterized chiefly by one of four fundamental features, which formed a basis for dividing the controls into the following general types:

Type I: Automatic control of furnace damper operation only, with the fan operating continuously and independently of any control. (Control *B.*)

Type II: Automatic control of both the fan operation and the furnace damper operation; so that the fan operated intermittently, and the damper-open periods occurred irrespective of whether or not the fan was running. (Controls *A, D,* and *I.*)

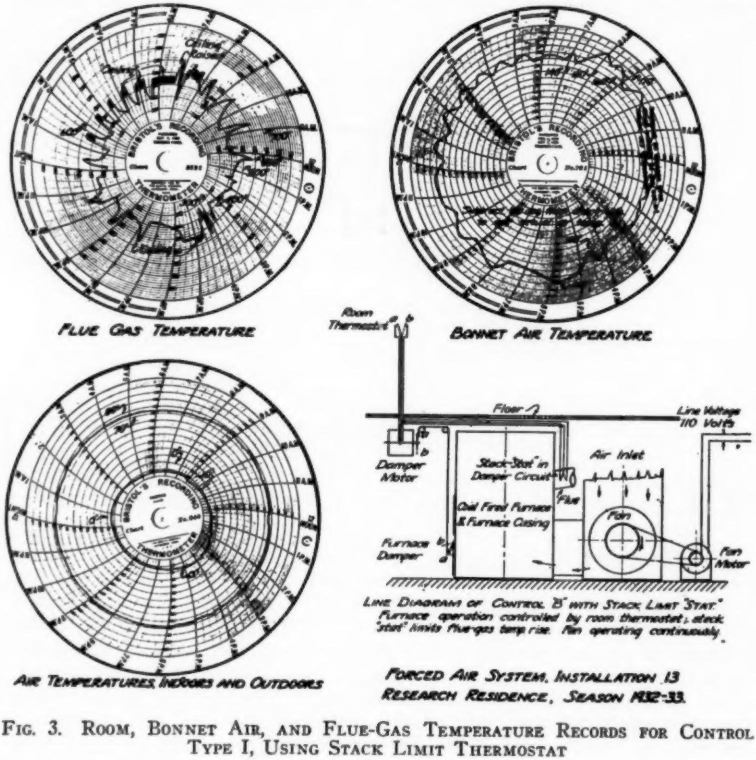


FIG. 3. ROOM, BONNET AIR, AND FLUE-GAS TEMPERATURE RECORDS FOR CONTROL TYPE I, USING STACK LIMIT THERMOSTAT

Type III: Automatic control of both the fan operation and the furnace damper operation; so that the fan operated intermittently, except during the damper-open periods when it was rendered totally inoperative. (Control *E.*)

Type IV: Automatic control of both the fan operation and the furnace damper operation; so that the fan operated intermittently, and the damper-open periods accompanied either all, or some fraction of each of the fan-on periods. (Controls *C, F, G, H, J,* and *K.*)

In the case of Types II and IV the installations which gave the most satisfactory performance were controls *A* and *H* respectively. These have, therefore, been selected as representative of the two types, and in the following discussion only the representative controls have been considered in detail to illustrate the performance characteristics of the four fundamental types and reference to them has been made by type alone.

Type I. (Figs. 1, 2, and 3.)

In this case, the room thermostat operated the damper motor directly, and the fan was allowed to run continuously. This arrangement gave a control system that was very simple and dependable. See Fig. 2 for the wiring diagram.

The quantity of air delivered by the fan was practically independent of the bonnet air temperature; consequently, positive and continuous air circulation was maintained, and both the air stratification in the rooms and the resulting room air temperature gradients from floor to ceiling were minimized. Also, for a given adjustment of the volume dampers in the duct system, equal temperatures were found to exist in the different rooms of the house for wide ranges of outdoor weather conditions. The temperature charts in Fig. 1 indicate that the temperatures of the bonnet air and flue gas were very low. These low temperatures were conducive to a condition of maximum bonnet efficiency and, in view of minimum heat losses from the ducts, of maximum register efficiency. Since the air movement through the register oulets was constant, the register air temperatures were necessarily reduced appreciably in very mild weather, causing perceptible drafts in the vicinity of some of the register outlets.

The temperature charts in Fig. 1 indicate further that the flue gas and room air temperatures tended to over-run slightly after each opening of the furnace damper, but the over-runs were not serious if the room thermostat was sensitive.

The use of a stack limit thermostat, placed in the flue, to close the furnace damper when the flue gas temperature rose to a predetermined value tended to minimize all temperature over-runs. The operating charts in Fig. 3, for a very severe day, illustrate the results obtained with the stack limit thermostat when used in this connection to prevent the flue gas temperature from exceeding approximately 600 F. The fan was operating continuously, and when the room thermostat demanded heat the damper opened. However, due to the use of the stack limit thermostat, which was set for a temperature high enough to handle the severest weather conditions, a definite *ceiling* was placed on the flue gas temperature. The stack limit thermostat, when used in this connection as an integral part of the control system, was found to be most effective in preventing temperature over-runs, when it was adjusted to conform to the varying weather conditions. Its use in any manner, however, was not absolutely necessary for the successful operation of this type of control system.

Due to the high cost of fan operation with continuous drive systems, the use of controls with intermittent fan operation has been common in connection with forced-air heating systems. The three types of systems to be discussed in the following sections were in every case featured by intermittent fan operation.

Type II. (Figs. 4 and 5.)

This simply arranged control system consisted of a room thermostat which controlled the operation of the damper motor alone, and an auxiliary thermostat, located in the furnace bonnet, which controlled the operation of the fan

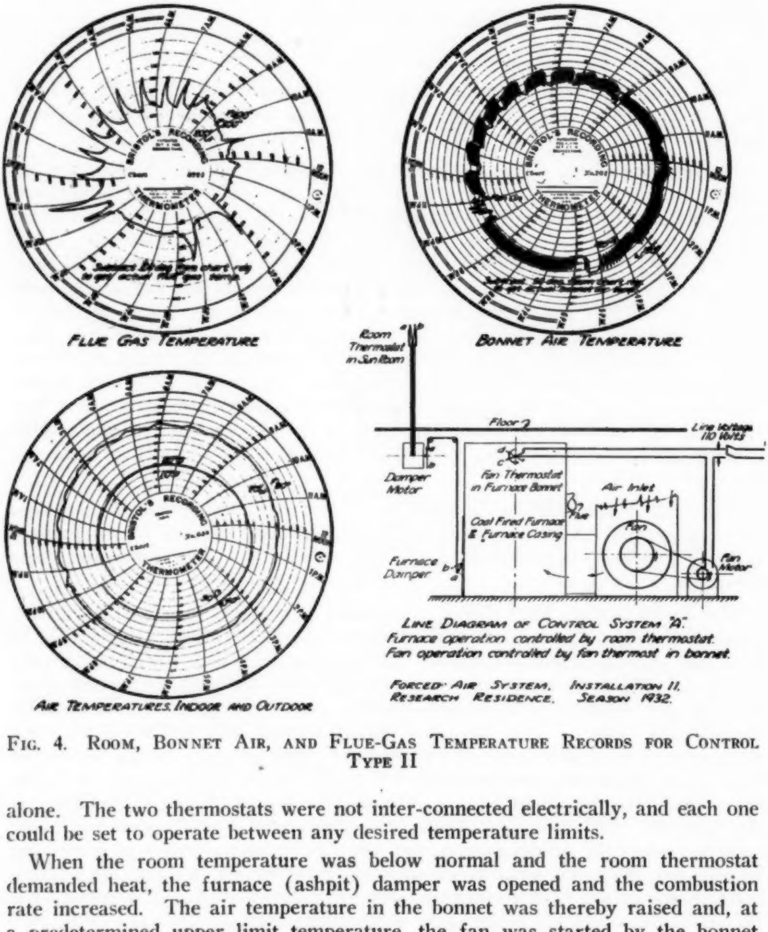


FIG. 4. ROOM, BONNET AIR, AND FLUE-GAS TEMPERATURE RECORDS FOR CONTROL TYPE II

alone. The two thermostats were not inter-connected electrically, and each one could be set to operate between any desired temperature limits.

When the room temperature was below normal and the room thermostat demanded heat, the furnace (ashpit) damper was opened and the combustion rate increased. The air temperature in the bonnet was thereby raised and, at a predetermined upper limit temperature, the fan was started by the bonnet thermostat. The reserve of heated air in the bonnet was soon forced out and the bonnet air temperature was reduced to the predetermined lower limit, depending on the setting of the bonnet thermostat, and the fan was stopped. The periods of *gravity* operation, intervening between the periods of fan operation, were characterized by an increase in the bonnet air temperature. This

cyclical operation of the fan continued as long as there was sufficient heat in the bonnet, irrespective of whether the room temperature rose to normal or not. The resulting tendency of the room air temperature to over-run the desired temperature of 72 F was due to this latter effect and is clearly shown in Fig. 4. The temperature chart for the bonnet air shows the very regular, cyclical operation of the fan between two set limits. The sequence of operations is shown in Fig. 5.

With the use of control Type II and for a given setting of the dampers in the duct system, unequal temperatures were obtained in the different rooms of the house during extreme weather conditions. This condition of unbalance was not serious in mild weather, but in very severe weather, the flue gas

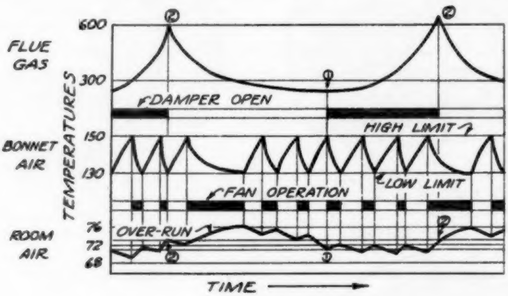


FIG. 5. SEQUENCE OF EVENTS FOR CONTROL TYPE II

temperatures were high and were accompanied by long periods of fan operation. In extreme cold weather operation, therefore, when the fan tended to run a major proportion of the day, the first story rooms became overheated, and the third story rooms were underheated.

Type III. (Figs. 6 and 7.)

In the case of control systems of both Type I and Type II, the fan operation was independent of the room thermostat. In the case of control Type III, however, the operation of the fan was controlled by the room thermostat through a fan relay switch, and the furnace damper was operated by a bonnet thermostat. The bonnet thermostat functioned in such a manner that the fan was rendered totally inoperative throughout the periods that the damper was open. The addition of the fan relay switch, together with a more elaborate bonnet thermostat, complicated the control system, but control Type III was as simple as most of the controls using a fan relay switch.

A block diagram accompanies the temperature charts in Fig. 6, and with the aid of this diagram, the cycle of operations may be easily followed from the temperature charts. During the period between 1:00 p. m. and 1:15 p. m., the damper was open for the entire time required for the bonnet air temperature to rise through a temperature differential of approximately 40 F, which corresponds to the differential between the lower and upper limit settings of the bonnet thermostat. During this period, the flue gas temperature rose to a value

in excess of 600 F, and the damper was not opened again until two hours later. It would seem logical to assume that if the original setting of the bonnet thermostat had been made so that the furnace damper remained open no longer than the time required to permit the bonnet air temperature to rise possibly only 15 F instead of 40 F, the combustion rate would have been maintained within narrower limits. A bonnet thermostat with this 15 F differential setting

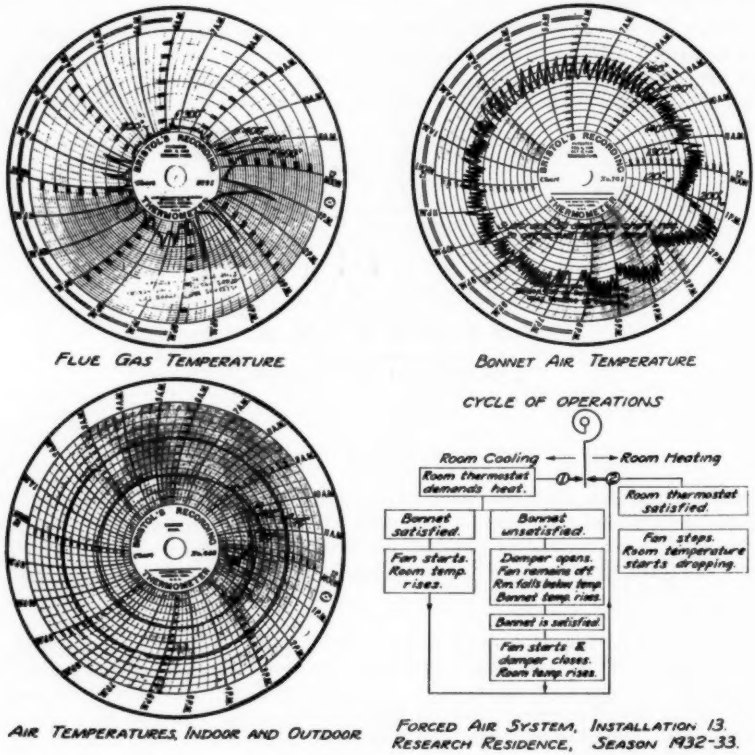


FIG. 6. ROOM, BONNET AIR, AND FLUE-GAS TEMPERATURE RECORDS FOR CONTROL TYPE III

was tested and the results were as anticipated. The furnace damper opened more frequently with this thermostat setting, resulting in a decrease in the magnitude of the over-runs in the bonnet air temperature.

Control Type III gave satisfactory operation provided that the setting of the bonnet thermostat was adapted to the heating demands being made on the furnace due to weather changes, and provided that the pick-up in the combustion rate was rapid. When the bonnet thermostat was adjusted for average weather conditions and when the adjustment was left unchanged in milder

weather, large over-runs in flue gas and bonnet air temperatures resulted. With colder weather conditions, however, the combustion rate was held below the value necessary to provide for the heating requirements.

Also, for cases in which the pick-up in combustion rate was slow, the room air temperature tended to drop four or five degrees below normal, since the fan was inoperative throughout the pick-up period. This condition is illustrated by Fig. 7. When the bonnet air temperature reached 120 F, as shown by point *3*, the fan stopped and the damper opened. The room air temperature fell from 71 F to 67 F as shown by points *1* and *2*, during the time that the bonnet air temperature rose from 120 F to 160 F, as shown by points *3* and *4*. Here again a bonnet thermostat differential, *d*, of 15 F would give better results than those given by a differential of 40 F.

This system was adapted to handle very sudden heating demands, as in morning pick-up periods, without allowing the furnace damper to remain open long enough to dangerously overheat the furnace. Since the fan was never

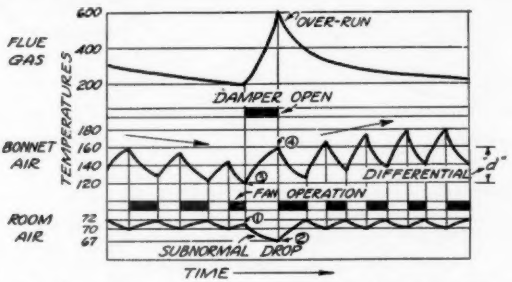


FIG. 7. SEQUENCE OF EVENTS FOR CONTROL TYPE III

running after the furnace damper had been opened, the bonnet air temperature rose very rapidly to the value at which the bonnet thermostat closed the furnace damper.

Type IV. (Figs. 8, 9 and 10.)

The results obtained with controls of the types discussed in the foregoing sections had shown that whenever the heating requirements of the house were supplied by short periods of rapid combustion followed by long periods of slow combustion, the former periods were followed by more or less serious over-runs in temperature either at the furnace bonnet or in the rooms. Since the furnace casting proper had relatively little capacity for storing heat, the heat that was evolved during the periods of rapid combustion was not held in reserve for gradual use but was rapidly absorbed by the air, with the result that temperature over-runs occurred.

However, the results obtained with a preliminary control arrangement had shown that when the furnace damper was opened frequently and for short periods of time, a much more uniform combustion rate was maintained. The actual operation of this preliminary control was equivalent to the operation of a control system by which the fan operation was controlled directly by the

room thermostat alone, and the damper was held open throughout each entire fan-on period. Two significant facts, which had an important bearing on the design of control Type IV, were established by the results obtained with the preliminary control. They were (1) that a uniform combustion rate just sufficient to supply the heating requirements, without great variation from a mean value, could be maintained by allowing the frequency of the damper-open

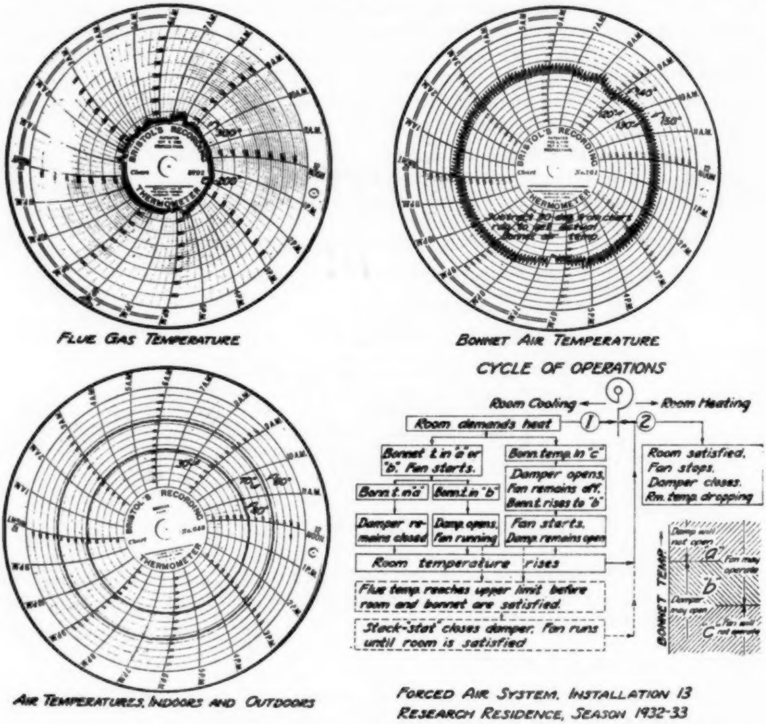


FIG. 8. ROOM, BONNET AIR, AND FLUE-GAS TEMPERATURE RECORDS FOR CONTROL TYPE IV

periods to correspond to that of the fan-on periods, and (2) that excessively high flue gas and bonnet air temperatures existed when the furnace damper was open throughout each fan-on period. It, therefore, seemed reasonable that a similar control system which would limit the high flue gas and bonnet air temperatures obtained with the preliminary arrangement would give very satisfactory results.

It was found that the temperatures could be most successfully limited by the use of a control arrangement that would allow the furnace damper to be open during only a part of each fan-on period rather than for the entire

period as in the preliminary arrangement. Control Type IV, therefore, was designed (1) to automatically open the furnace damper during each fan-on period, provided that the bonnet air temperature fell below a predetermined value and (2) to automatically adjust the length of time that the damper remained open during each cycle, proportional to the amount that the bonnet air temperature fell below the predetermined value. The first feature provided uniform combustion as a consequence of frequent damper openings, and the second feature maintained a uniform combustion rate just sufficient to supply the heating demand of the house. This latter result was accomplished in the following manner: As the heating demand increased, the heat in the bonnet air tended to become depleted and at the end of each fan cycle the bonnet air temperature assumed successively lower values as represented by points *3a, 3b,* and *3c* in Fig. 9. This downward trend caused the pick-up period, represented by the distance along the time axis between points *2'* and *3'* to be lengthened, with the result that the combustion rate, as indicated by the flue gas temperature, increased sufficiently to supply the increased heating demand.

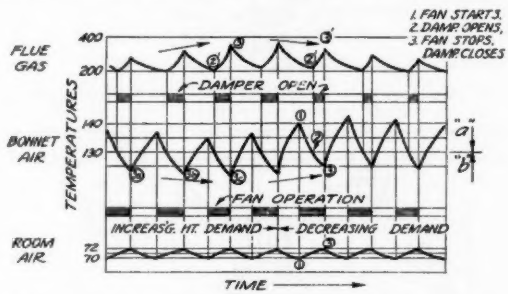


FIG. 9. SEQUENCE OF EVENTS FOR CONTROL TYPE IV

As the heating demand decreased, the reverse process took place, with the result that the combustion rate was decreased. Therefore, by making the frequency of damper opening dependent on the frequency of the fan cycle, and by making the length of time that the damper remained open dependent on the bonnet air temperature, a very satisfactory control of the temperatures throughout both the house and heating plant was maintained.

All of the above features were incorporated in control Type IV by using a room thermostat to control the operation of the fan through a fan relay, a bonnet thermostat to control the furnace damper as explained above, and a damper motor. This equipment was so interconnected electrically that the furnace damper could not open, unless the room thermostat was calling for heat, even though the bonnet thermostat was in the *damper open* position. The manner in which these connections were made is shown by the wiring diagrams in Figs. 10, *a, b.*

The temperature records and the block diagram indicating the cycle of operations for control Type IV are shown in Fig. 8. The combustion rate as indicated by the flue gas temperature varied only slightly from a mean value that was just sufficient to supply the heating demands being made on the

plant. With only one setting of the bonnet thermostat, equal temperatures were obtained in the different rooms of the house for a wide range of weather conditions. The most satisfactory operation was obtained, however, when the bonnet thermostat setting was adjusted to two values—one value for mild weather and a higher value for severe weather.

In the dotted line portion of the block diagram, mention is made of a stack limit thermostat, which during normal operation did not enter into the operating cycle, except in very severe weather. Its purpose was to limit flue gas tempera-

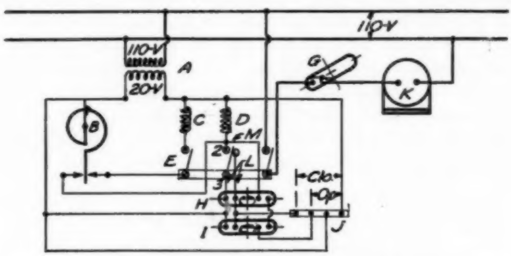


FIG. 10*a*. CONTROL TYPE IV (TWO-POINT THERMOSTAT)

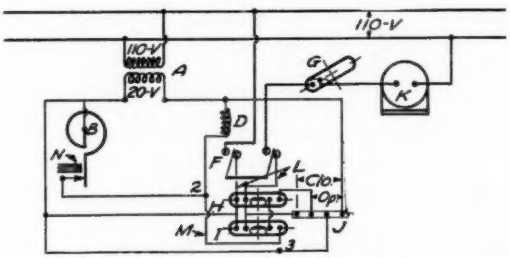


FIG. 10*b*. CONTROL TYPE IV (ONE-POINT THERMOSTAT)

A—transformer; *B*—room thermostat; *C*—bucking coil (relay out); *D*—pulling coil (relay in); *E*—4-circuit relay switch; *F*—2-circuit relay switch; *G*—fan thermostat; *H*—bonnet thermostat; *I*—stack limit thermostat; *J*—furnace damper motor; *K*—fan motor; *L*—aux. damper closing circuit; *M*—damper opening circuit; *N*—permanent magnet.
Notes: For Control *F*—omit auxiliary closing circuit marked (*L*). For Control *J*—omit aux. closing circuit (*L*) and change connection of Circuit (*M*) from point (2) to (3).

tures when sudden heating demands, resulting from setting the room thermostat for a higher temperature, were made on the plant. When used in this connection, the stack limit may prove of value not only from the standpoint of safety but from the standpoint of economy as well. Previous tests (see *Bulletin 246,* University of Illinois Engineering Experiment Station, pages 96-101) have shown that the flue gas losses incurred during heavy morning pick-up periods tend very materially to offset the savings resulting from lowering the house temperatures at night. The use of a stack limit thermostat would minimize such flue gas losses.

COMPARISON OF CONTROL SYSTEMS

To a certain extent the power consumption of the fan, as measured by the power input to the fan motor, was influenced by the range of bonnet air temperature selected for setting the bonnet thermostat and by the condition of the fuel bed. Both of these factors had some influence on the number of times that the fan was required to operate during one or several cycles of events. As a rule, the condition of the fuel bed was not directly related to the type of control and the effect was a random variable as far as the type of control was concerned. The selection of the range of bonnet air temperature to be used in setting the bonnet thermostat was in some cases dictated by the operating characteristics of the control, and in some cases dependent on the judgment of the operator. In the latter cases it was sometimes a matter of choice between using a range that would permit fairly satisfactory operation over the whole range of weather conditions, and using several ranges of bonnet air temperature best adapted to narrow ranges of weather conditions and changing the setting as the weather varied. All tests were run under conditions considered most reasonably adapted to the control in use, but occasional inconsistencies in power consumption appeared, and it is well to emphasize that in individual cases in practical service the power consumption might deviate considerably from the results obtained in these tests.

The power consumption of the fan in kilowatt-hours per day for various indoor-outdoor temperature differences is shown in Fig. 11-*a*. Some deviations of the points from the curves may be explained by the fact that some variations in the heat load on the house, caused by wind, sun, and rain, can not be expressed directly in terms of indoor-outdoor temperature differences. Most of the deviations, however, were the result of the factors discussed in the preceding paragraph. Hence, the trend of the points has been represented by a mean curve, and by two boundary curves indicating the probable range of deviation attributable to variations in the condition of the fuel bed and in the setting of the bonnet thermostat. With these restrictions, it may be observed that for intermittent operation, the power consumption of the fan motor was more or less independent of the type of control and increased as the indoor-outdoor temperature difference increased. This increase was from approximately 0.4 kilowatt-hours per day at 20 F difference to approximately 3.0 kilowatt-hours per day at 60 F difference. On the other hand, the power consumption for continuous fan operation with control Type I was not greatly influenced by the random factors mentioned and remained practically constant at 3.6 kilowatt-hours per day over the whole range of indoor-outdoor temperature differences. It may be further observed that the power requirements for both continuous and intermittent operation became identical at an indoor-outdoor temperature difference of about 64 F.

The total number of hours that the fan was actually running during a 24 hour test period, as determined by means of a self-starting electric clock included in the fan circuit, is shown in Fig. 11-*b*. At an indoor-outdoor temperature difference of 64 F, at which the power requirements for continuous and intermittent operation became the same, as shown in Fig. 11-*a*, the total number of hours of operation per day was 18 for intermittent operation as compared with 24 for continuous operation. This indicated that for the same number of hours of actual running the fan motor consumed more power

when running intermittently than when running continuously. This effect has been attributed to the extra load required to start the motor.

The amount of the extra load imposed by starting the motor is shown in Table 1. The kilowatt-hours per hour of actual running time, was obtained by dividing the kilowatt-hours per day shown in Fig. 11-*a* by the total number of hours of running time shown by Fig. 11-*b*, at the corresponding indoor-

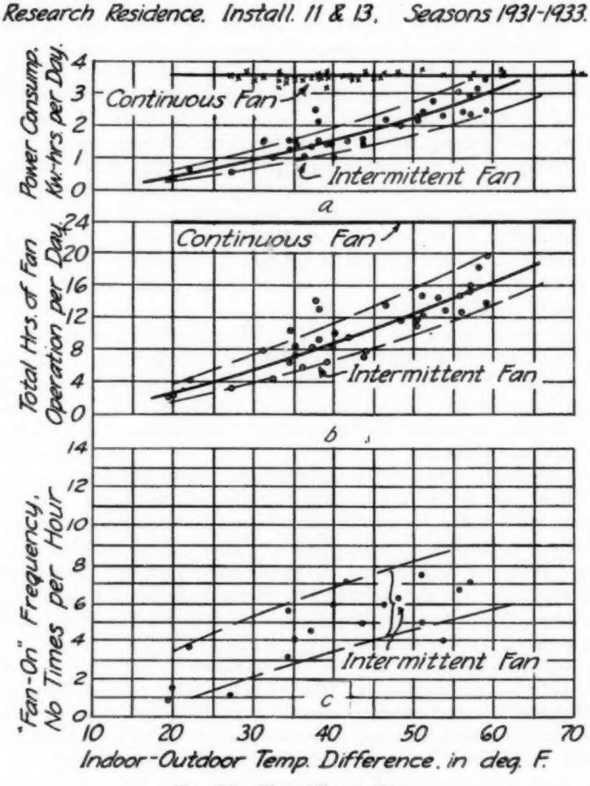


FIG. 11. FAN MOTOR DATA

outdoor temperature difference. For continuous operation this remained constant at a value of 0.150.

From Table 1 it is evident that in average weather, or at an indoor-outdoor temperature difference of 40 F, the power requirement per hour of running time was 18 per cent greater for intermittent operation than it was for continuous operation. In severe weather, or at an indoor-outdoor temperature difference of 64 F, the starting load constituted an additional load of approximately

30 per cent of the normal running load. The power for starting is influenced by the type of motor and it is possible that if a split-phase or capacitor motor had been used the power for starting would have been greater.

Fig. 11-*c* shows the average number of times the fan was operated per hour in order to maintain the house temperatures, as well as the increase in the number of cycles as the load demand was increased. Under conditions of normal heating demand of the house it was found that a frequency of *fan-on* cycles of between four and eight times per hour gave satisfactory performance. Higher frequencies would tend to give closer regulation of the room temperature but the numerous starting-up periods might be detrimental to the fan and motor equipment. On the other hand, frequencies less than four times per hour in normal heating weather did not give satisfactory regulation and equalization of room temperatures.

In all cases of intermittent fan operation, the temperature gradients from floor to ceiling were larger than those obtained in the case of continuous fan

TABLE 1. POWER CONSUMPTION PER HOUR OF ACTUAL RUNNING TIME

Indoor-Outdoor Temperature Difference Deg. F.	Kilowatt-Hours per Hour		Increase Due to Starting Load Per Cent
	Continuous Fan Operation	Intermittent Fan Operation	
20	0.150	0.155	3
40	0.150	0.177	18
60	0.150	0.190	27
64	0.150	0.196	30

operation. During these tests, however, there was no noticeable tendency toward disagreeable air stratification in the rooms.

The results from two years of operation in the Research Residence have shown that there was no marked difference in fuel consumption, or overall house efficiency, with operating methods that varied as widely as the difference between all-gravity and all-fan operation, and further that the fuel consumption was practically independent of the type of automatic control system used. Although the efficiency, as determined by the heat delivered at the furnace bonnet and warm air registers was higher for the all-fan operation, the overall house efficiency, due to the reabsorption into the house of the stray heat losses from casing, ducts, smoke-pipe, and chimney, was not markedly different from that obtained with other methods of operation.

Control Type I had many merits, and in some cases, particularly in large installations, the advantages of continuous fan operation far outweigh the disadvantages. Four suggestions are offered which would overcome to some extent the chief disadvantages of high cost of continuous fan operation and of low register air temperatures in mild weather:

1. By using suitable diverters in the register outlets, thus tending to spread the air stream and reduce the velocity, or by using high sidewall registers, the flow of air from the registers may be made unobjectionable even for low air temperatures at the register outlet.

2. By providing for an auxiliary fan thermostat in the furnace bonnet, as indicated in the wiring diagram shown in Fig. 2, the fan may be stopped when the bonnet air temperature becomes objectionably low.
3. By suitable design, the plant may be made to operate on register air temperatures of from 160 F to 175 F instead of the 135 F now commonly used, thus permitting the use of a fan with smaller air capacity.
4. By using a two-speed drive on the fan motor, the fan speeds can be regulated to maintain a more uniform register air temperature for all heating loads. The low speed drive would be used for possibly four-fifths of the heating season, and the high speed drive for extremely cold weather only.

With continuous fan operation, satisfactory performance was obtained by means of the simple control equipment commonly used in gravity installations—a room thermostat and a damper motor. With intermittent fan operation, however, the minimum of equipment that could be used included a bonnet thermostat in addition to the room thermostat and damper motor. This minimum of equipment was used in the case of control Type II and the results obtained were not satisfactory, as indicated in the foregoing discussion.

The results obtained with controls Type III and Type IV, on the other hand, show that satisfactory conditions may be maintained in the house with the intermittent fan operation provided that a fan relay switch is incorporated in the control system so that the fan operation may be governed by the room thermostat. The partly satisfactory results obtained with control Type III show that with this combination, frequent adjustment of the bonnet thermostat was necessary to prevent temperature over-runs. Control Type III had one advantage over control Type IV in that when sudden demands were made on the plant, the combustion rate was successfully limited. This advantage, however, was offset by the fact that throughout each pick-up period the fan was totally inoperative, and as a consequence the room temperature fell below normal during these periods.

The very satisfactory results obtained with control Type IV show that it was adaptable to normal operation over a wide range of weather conditions without changes in the initial adjustment. Also, the combustion rate could be successfully limited at all times by the addition of a stack limit thermostat, although it did not become excessively high even without the use of a stack limit. The results obtained with the use of control Type IV, therefore, show that it was possible to obtain results with intermittent fan operation that were as equally satisfactory as those obtained with continuous fan operation without involving the high operating costs characteristic of the latter arrangement.

CONCLUSIONS

The following conclusions may be drawn as applicable to the conditions under which the tests were conducted:

1. The room air temperature may best be maintained within definite limits by operating the fan from the room thermostat (through a fan relay).
2. For ideal operation, the operating range of bonnet air temperatures should be governed either manually or automatically, to correspond to the heating demands. A system which tends to maintain a fixed range of bonnet air temperature may be in adjustment to give equal temperatures in the different rooms for a fairly wide range of weather conditions, but for extreme weather conditions this adjustment may result in unequal temperatures in the different rooms.

3. The most satisfactory operation of the heating plant required the maintenance of a uniform combustion rate just sufficient to supply the heating demands being made on the furnace. This requirement involves frequent openings and closings of the furnace damper, and can best be fulfilled when the damper operation is dependent on both the room temperature and the bonnet air temperature. This is equally true for gravity and forced air systems.
4. In systems using intermittent fan operation, uniformity in the length of the cyclical periods of the fan is necessary in order to maintain equal temperatures in the different rooms. Excessively long periods of either fan operation or *gravity* operation may cause overheating in some section of the house apart from the control room.
5. With control systems involving intermittent operation of the fan, the fan should be started between four and eight times per hour in order to obtain satisfactory regulation and equalization of temperatures in the different rooms.
6. Continuous operation of the fan has many advantages, particularly in large installations.
7. The total power required in the case of intermittent fan operation varies from 10 per cent to 100 per cent of the power required for continuous fan operation. In the case of intermittent operation, the power required to start the fan contributes an additional load varying from 3 to 27 per cent in excess of the normal running load, over a range of indoor-outdoor temperature differences of from 20 F to 60 F.
8. The fuel consumption and overall house efficiency for intermittent fan operation and continuous fan operation are practically the same.
9. A stack limit thermostat arranged to prevent flue gas temperatures from exceeding a predetermined upper limit can be successfully used to limit the combustion rate during sudden load demands.

DISCUSSION

PERRY WEST: Does the over-all efficiency include the cost of current as well as the cost of fuel? As I understand it, if the authors mean by efficiency the total over-all cost of operation, the cost was the same in each case.

E. K. CAMPBELL: To what extent is the result of this test applicable to bituminous coal?

W. W. TIMMIS: Did the authors make any observations of the relative efficiencies of combustion with the short periods of on and off operation of the furnace damper, as compared with the less frequent but longer periods of furnace damper operation?

C. H. TURLAND: Was an analysis of the flue gases taken over a 24-hour period at different times?

ROSWELL FARNHAM: What is the difference in cost of continuous fan operation in electrical input against the intermittent?

S. KONZO: The first question raised was in regard to the over-all efficiency of the house. This efficiency refers only to the ratio of the amount of heat absorbed by the house to the amount of heat generated in the furnace, and therefore does not take into account the electrical costs. Since the over-all efficiencies for the two cases of gravity and forced-air systems were approximately the same, the electrical cost of fan operation would tend to make all of the forced-air systems a little more expensive to operate than gravity systems.

Mr. Campbell raised the question as to whether the results are applicable to bituminous coal. In addition to anthracite coal, both bituminous coal and coke were burned and it was found that the operation of control Type IV was very satisfactory with these fuels.

Mr. Timmis raised the question of relative efficiency of short periods of damper operation versus long periods of damper operation. The CO_2 records for the two cases show that with very short damper periods, that is for conditions where a uniform combustion rate is maintained, the CO_2 is maintained at a fairly high, uniform value, whereas with long periods of damper operation, the fluctuation between the maximum and minimum value of CO_2 is very large. In the latter case, during the periods of high CO_2 values the volume of flue gas generated is large. Consequently, the periods of large generation of heat coincide with periods of high combustion efficiency. Conversely, when the heat generation is a minimum, the combustion efficiency is low. An average combustion efficiency that is based on an arithmetical average of the CO_2 value is thus meaningless under these conditions. However, the evidence presented from the fuel consumption for the two cases points to the conclusion that whatever difference may exist in the relative efficiency of short damper periods, as compared with long damper periods, it is so small as to be indistinguishable in an installation such as in the Research Residence, where the heat losses from the plant are absorbed into the structure.

The question by Mr. Turland was in regard to flue gas analysis. These were not taken on all tests but were noted on a sufficient number to indicate that combustion was quite satisfactory in the case of control Type Four and was less satisfactory in the case of the other controls in which the damper was not operated very frequently. Here again the same factors apply that were stated in the previous answer.

The last question was in regard to the electrical cost of intermittent operation as compared with continuous fan operation. The continuous fan operation, as we have indicated, consumed about 3.6 kwhr for the 24-hour period for this particular installation. The cost of intermittent operation amounted to a value anywhere from 10 to 100 per cent of that for continuous operation.

MR. FARNHAM: The item of starting torque was interesting. How important a part does that play?

MR. KONZO: As indicated in the paper, the power consumed in starting the fan in this installation varied from 3 per cent of the normal running load to 30 per cent, depending on how often the fan was operated. The value of 30 per cent holds for extremely cold weather conditions.

It must be remembered that all of these tests were taken with a particular type of motor, the repulsion-induction type. It is possible that with other types of motors this starting load may be higher.

No. 971

HEATING BUILDINGS WITH HOT WATER

By Benj. F. Burt * (*NON-MEMBER*), Rochester, N. Y., and Samuel R. Lewis † (*MEMBER*), Chicago, Ill.

IN the days when many good residences were heated by hot water under gravity, or thermal circulation, the systems gave excellent satisfaction, circulation to each radiator usually being even and the heat delivered to the rooms being proportional to the intensity of the fire in the boiler. Occasionally buildings are encountered today, where the hot water heating system is upward of 50 years old, and is functioning as well as ever, with little or no repair expense during the years of service.

As civilization advanced there came a demand for a less expensive and less artful type of heating and the use of hot water circulation was neglected. Single-pipe or air vent low pressure steam heating became popular. It was more easily installed than thermally circulated hot water heating and as the boiling point of water was the lower temperature limit of the steam system, radiators smaller than those needed with hot water could be used. In the early systems the radiators were filled with steam in mild weather, and rooms were overheated; consequently windows were opened and heat was wasted. No graduated regulation of temperature by valves was possible.

People therefore hailed the advent of the patented vacuum method of double pipe steam heating. It did away with the hissing air valves of the single pipe systems. One could, if the radiators had both top and bottom connections between the sections, regulate the room temperature by valves. If the piping and the valve packing were tight the temperature of the evaporated water inside the pipes and radiators could be reduced from the 215 F of the single pipe steam system to perhaps 160 F by maintaining a pressure below atmospheric. This pressure and temperature, if in effect on the water surrounding the fire in the boiler, would promote fuel economy, since the cooler the heat absorbing material the cooler the flue gases passing up the chimney.

The saving in fuel made possible by reduced temperature of the heating medium in high vacuum heating systems has been demonstrated thoroughly. It may be accounted for to some extent by the fact that there is less opening of windows in order to cool off. The difficulty and cost of keeping these high vacuum systems in perfect physical condition year after year to maintain the high vacuum is also very real.

---

* Vice-President, Rochester Circulator Corp.
† Consulting Engineer.
Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by H. L. Alt.

Vacuum systems of steam heating using pumps, and their cousins, the vapor systems without pumps, were promoted widely and the patented steam specialties used in these systems were extensively used.

In many cases with steam heating, complicated and elaborate piping systems and special building construction were necessary to protect the return pipes.

People still were neglecting hot water heating, which could provide a wide water temperature range and permit graduated room temperature control. Hot water heating may have a temperature range from room temperature to 250 F or so, and needs only two main pipes. When a circulating pump is used, all radiators, whether below or above the mains, may be served equally well.

During these many years the idea which would bring hot water heating back to serve civilization was alive but decidedly dormant. A few people remembered it and cherished it. Hydraulic forced circulation is, of course, nature's method of heating humans and animals. Thermal or gravity circulation of the blood if ever tried was abandoned by nature. Possibly thermal circulation is approached by the sluggish type of reptile during the hibernation period. The hands and feet of these reptiles get exceedingly cold during their quiescent times when the pumps are slowed down.

In man the heart is the pump for his hydraulic heating system and gives positive circulation so that if all is well the right hand is as warm as the left hand. So in hot water heating with mechanical circulation the radiator most remote from the boiler easily can be compelled to maintain practically the same mean effective temperature as that of the nearest radiator.

Nature couldn't use a rotating pump, but apparently was forced to limit her service to a reciprocating design. Man never has been able to design a reciprocating pump which approached the probable efficiency of the heart pump designed by nature, but man has the wheel and its derivative, the rotary or centrifugal pump. Man plus the lubricated wheel and its family comes very near to surpassing nature in mechanical efficiency.

With a power driven fan applied to an air stream, man can blow air far less wastefully than he can blow it by using a fire at the bottom of a chimney. With a power driven pump applied to a water stream man can circulate water far more efficiently than he can circulate it by applying heat to part of it to reduce its weight per unit of volume, so that the cooler water will fall down below the warmer water and thus start circulation.

The heat demand for every building changes constantly. At noon it may be rather light, due to occupancy by people, to work, or to the sun. At evening the heat demand may increase because of the slowing down of activities and absence of sunshine. At night the demand may decrease or cease for a time due to a lower temperature requirement. In the morning the heat demand may be very heavy to regain the night loss, since the desire will be for a higher temperature than during the night, and the sun has not yet had time for action. All of these changes may be upset by storm or by sudden changes in the heat production from people, lights or machines. In addition to daily, or rather local, changes in temperature, the heating system must also meet the general seasonal changes. The heating season in most localities begins with an occasional slight demand for heat in September and ends with a similar demand in May. There is the heavy and constant requirement of heat in January and the varying and

often widely fluctuating needs of the intermediate months. Thus to be comfortable and economical any heating system must be adjusting itself constantly to the conditions outside and inside, and if it can anticipate these conditions it will be a still better heating plant.

The smaller thermally circulated hot water heating systems accomplish this rapid adjustment. The very large thermal water heating systems are often sluggish and fairly subject to criticism. The single pipe steam systems and the vacuum steam systems gave potential quick response to the fire intensity. Any heating system which overshoots the mark will cause windows to go up to prevent overheating and will be wasteful of fuel.

It must be remembered that each room must have enough radiation to heat the space in the coldest weather. If steam is used a temperature of around 212 F will be necessary, and each radiator usually will be hot all over if hot at all. With controlled high vacuum this temperature may be reduced considerably. In mild weather with steam heating there is an almost irresistible tendency to overheat in flashes and to waste fuel by opening windows.

The response of mechanically circulated hot water heating systems to the change in heat demand by the room is as quick as the response with steam, but the controlled water temperature on a mild day may be as cool as 100 F or 120 F and the room temperature will not over-run, the windows will not be opened to relieve overheating, and much fuel will be saved.

Where old thermally circulated hot water heating systems have had faults due to inferior design or workmanship, a mechanical circulator easily can be added, with the almost invariable result that the sluggish circuits or radiators are corrected, and with the equally interesting results that fuel savings are registered.

With mechanical water circulation the water temperature difference between radiator inlet and outlet may be varied within a very wide range and the circulation rate is independent of water temperature. With thermal circulation, to produce flow there must be a considerable temperature difference between flow and return water. If there is one unfavorable circuit or radiator, all the other circuits or radiators must pass an unnecessary volume of water or must be throttled so as to carry a greater temperature difference between inlet and outlet than the sluggish circuit, so as to force some action out of the latter.

Consider water at 180 deg *flow* temperature circulating to a radiator in a room during a period of maximum demand, and that this maximum demand rate is, for this room, 4500 Btu per hour (4.5 *Mbh*). Assume that the radiator in this room, receiving water at 180 F, will give off the necessary heat (approximately) at a mean effective temperature of 170 F, and the water will leave the radiator at 160 F. There will be a temperature drop of 20 deg; $\frac{4500}{20} = 225$ lb of water to be circulated each hour (7.5 lb per 150 Btu h or 50 lb per *Mbh*). This gives no consideration to the amount of radiator surface, which of course must be adequate, and makes no allowance for losses from the piping.

A drop in water temperature of 20 deg is the customary design allowance from the boiler outlet back to the boiler inlet, but in practice some thermally circulated radiators may have a temperature drop of only 1 or 2 deg while others have such resistance to flow as to require a temperature drop of 30 deg

or more to develop the necessary power to move sufficient water through them to deliver the desired heat output in Btu per hour, even at this greater temperature drop. The boiler, which must develop this power, often is handicapped by the too-rapid thermally induced flow through the more favorable circuits. With milder weather and need for water at a reduced temperature in systems like this, the unfavorable circuits consequently become more and more sluggish as the available temperature head decreases.

Mechanical circulation of the water corrects these difficulties and permits or compels a balanced circulation, with a maximum and assured temperature drop

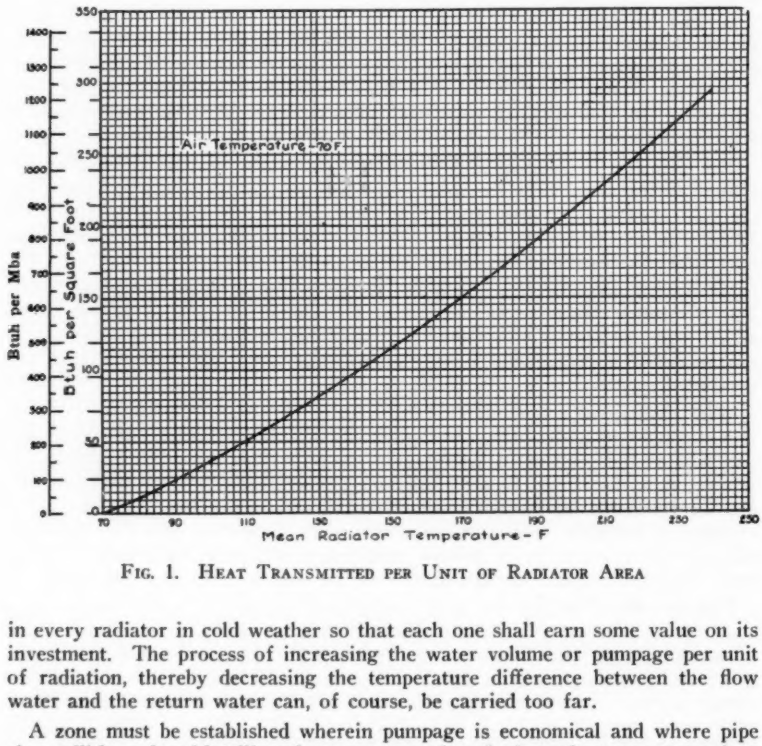


FIG. 1. HEAT TRANSMITTED PER UNIT OF RADIATOR AREA

in every radiator in cold weather so that each one shall earn some value on its investment. The process of increasing the water volume or pumpage per unit of radiation, thereby decreasing the temperature difference between the flow water and the return water can, of course, be carried too far.

A zone must be established wherein pumpage is economical and where pipe sizes will be reduced intelligently to correspond, and where the temperature drop through the boiler and radiators gives a rationally balanced efficiency. Herein appears the engineering skill and experience required for design of hot water heating.

The amount of heat transmitting surface required in any room is a function of the temperature difference between the air in the room and the mean effective temperature of the circulating medium within the radiator. The amount of heat transmitted per unit of radiator area touched by the air varies

with the shape and height of the area, and with the speed at which the air passes over it.

In discussing units of heat transmitting surface there always is complication when the obsolescent term square foot is used. One wonders always whether it means some unit which transmits 240 Btu h when in air at 70 F or whether it means the same area transmitting 150 Btu h. The new term *Mb* proposed by the Committee on Nomenclature of the Society means 1000 Btu. *Mbh* means 1000 Btu per hour. *Mba* means the ***area*** which at 215 deg mean temperature of the heating medium transmits 1000 Btu per hour, in the standard room temperature of 70 F.

When water is used as the heating medium a unit area which would transmit 1000 Btu h (1 *Mbh*) when full of steam at 215 F in air at 70 F will transmit less heat or more heat, depending on the water temperature. Throughout this paper, therefore, values will be expressed in both *Mb's* and in square feet of radiation, the *Mb's* usually being the number of heat units per hour transmitted at the stated water temperatures by the same surface area which would transmit 1 *Mb* if the water were at 215 F in thermally circulated air at 70 F. Given a room which loses 10 *Mbh* to be heated by a water radiator under conditions in which each unit of surface transmits 0.6 *Mbh* there would be required $\frac{10}{0.6}$ or 16.7 *Mba* of radiator surface area.

With a mechanically circulated liquid any two units of area under the same conditions for both will give off heat proportionally as their mean effective temperatures vary.

With hot water the given mean effective temperature may be varied within a very wide range, extending from the room temperature itself well beyond the temperatures of low pressure steam. The heat transmitted to the room per unit of radiator area may likewise be varied in a complete and unbroken range, from no heat at all up to the maximum required. The accompanying chart visualizes the extent of this possible range. It agrees closely with the data given in THE A. S. H. V. E. GUIDE 1933.

It would be possible to change the temperature of the entering water and of the outgoing water within a very wide range indeed without increasing or decreasing the amount of necessary radiating surface, so long as the mean effective temperature is kept constant. Thus for a mean effective temperature of 170 deg, commonly used as a basis for design:

Temperature of Entering Water F	Temperature of Leaving Water F	Temperature Drop in Radiator F	Mean Effective Temperature F
175	165	10	170
180	160	20	170
185	155	30	170
190	150	40	170
200	140	60	170

When steam is used inside a radiator the pumpage phase of the matter is of little moment. The radiator is kept full of steam which condenses to water

that occupies only about one-eightieth of the space of the steam from which it was condensed. With a liquid inside the radiator the volume does not change appreciably as the temperature changes. After heating the outside of the radiator nearly to the temperature of the liquid nothing is to be gained by forcing surplus liquid through the radiator. If one pumped enough water through the radiator he could reduce the temperature drop to a very low value and with the same entering temperature could increase slightly the heat transmitted per unit of radiation. The increase, however, would be by no means proportional to the pumpage.

For instance, with a mean effective temperature of 170 F and a 10 deg temperature drop the entering water would be at 175 F and the leaving water would be at 165 F. There would be required under these circumstances for delivery of 0.625 *Mbh*, $\frac{625}{10}$ lb of water per hour, that is, 62.5 lb of water per *Mba* of radiator area, or 15 lb of water per square foot of radiator area. With the same *entering* temperature as before, (175 F), doubling the pumpage to a

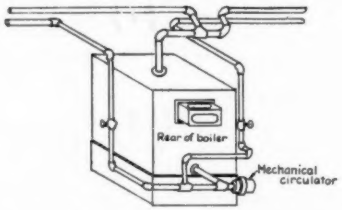


FIG. 2. TYPICAL RESIDENCE APPLICATION

figure of 125 lb of water per hour per *Mba* of radiator surface will raise the mean effective temperature of the radiator only 2.5 deg, and will increase the heat output of the radiator only about 0.020 *Mbh* per *Mba* unit area, or 4.75 Btu h square foot. This would be of very doubtful overall advantage, the only compensating factors against the greater cost of pumping being a very slight decrease in the necessary radiator size, and a slightly increased turbulence in the water around the heating surface of the boiler.

On the other hand, if the same *mean* temperature is maintained, that is 170 F, and if the pumpage rate is doubled to 125 lb per hour per *Mba*, the same heat delivery as before will be obtained, namely 0.625 *Mbh* per *Mba* unit area, and the only compensating factors against the higher pumping cost will be a slightly reduced temperature of the water around the fire and the same slightly increased turbulence or agitation of the water passing around the heating surface of the boiler.

Experience has demonstrated that in the design of new mechanically circulated hot water systems, using radiation of the same size as is customary with thermal circulation, and a mean effective radiator temperature of 170 F for the most extreme demand, a temperature drop of around 10 deg and a pumpage of about 62.5 lb of water per hour per *Mba*, or 15 lb of water per hour per 150 Btu unit of area, is conservative, and that it will give economical pipe sizes and power consumption.

If it is desired to use radiators no larger than those which would be required with steam, for hot water heating, the water temperature may be increased. Such increase in temperature of the water merely brings down the efficiency of heat transfer from fire to water more nearly to that in effect when steam is used. The saving in *first* cost due to smaller radiators must then be balanced against the increased *operating* cost due to the lower fuel efficiency, and the engineer must determine to which of these two factors he will give the greater weight. However, due to the very positive and rapid water travel against the heat absorbing surfaces around the fire with mechanical water circulation, it is believed that even at the same temperature differences the efficiency of a water boiler will be higher than that of a steam boiler.

The following explains one situation when higher water temperatures are used:

Pumpage per Hour		Ent. Temp.	Temp. Drop	Mean Eff. Temp.	Air Temp.	Conversion Factor	Btuh per Unit of Area	
Mba	Sq. Ft.						*Mba*	Sq. Ft.
118 lb.	28.4 lb.	240 F.	10 F.	235 F.	70 F.	1.183	1183	284

Throughout the milder weather which prevails most of the time during each heating season, a mechanical circulating plant will reduce the flow water temperature very greatly without prejudice to the distribution of heat, and will, unlike a thermally circulating plant, maintain this distribution while the flow water temperature is being reduced due to a reduction in the heat demand of the building.

During these hours the boiler water temperature often may be 30 deg or more below that required during those hours when the demand is increasing. The pump insures that all radiators are warm, but the comparatively cool boiler water, as well as a positive movement of the water across the boiler heating surfaces, gives a great increase in the efficiency of heat absorption from the gases of combustion.

This condition is in effect more than half of the time with any well designed mechanically circulated hot water heating system. The condition accounts for fuel savings inherent to a forced hydraulic heating system. For example in one large greenhouse the difference between the out-going water temperature and the return temperature was 90 F with thermal circulation. Thus the mean effective radiation temperature was 45 F below the radiator inlet temperature. Mechanical circulation reduced the 90 F drop to 10 F and maintained the same mean effective temperature with the same heat output, and accomplished saving in fuel which paid for the pump in a few months.

In greenhouses where Easter lilies are grown the flowers must be ready for the Easter trade exactly on time, or profits will be scant. Where hot water heat and thermal circulation are used the temperature of the house will be considerably cooler at the end served by the longest or most remote part of the water circuit. The plants respond to the warmer temperature so that if a string were stretched from the top of a lily at the cooler end of any row to the one

at the warmer end, the string would slant uphill sharply. This is of course a most undesirable condition. With adequate mechanical circulation having the temperature drop reduced to a negligible value the lily tops form a level line and all mature practically at the hour desired.

A new modern mechanical circulation piping system is very similar in general layout to a two-pipe vacuum steam system in that each radiator receives a supply connection from the flow system with a return connection to the separate return system. There are no connections from the supply mains to the return mains except through radiators, and the head which causes the water flow, positively is exerted on every radiator.

There are available automatic valves which may be placed on the various main circuits which will throttle the flow due to the pressure in the corresponding return, to which their pressure-sensitive elements are connected. Each automatic valve may be adjusted to permit the existence of the necessary pressure difference and no more, required to overcome the resistance of that particular circuit; thus making any surplus pressure available for use on the less favorable circuits.

The same automatic throttle valves may also be controlled by divisional thermostats so as to reduce the water flow as the room temperature increases. This permits the adjustment of the zone temperature automatically as the sun finds its way around the East, South and West sides of a building. In Northern cities the winter sun at noon looks far into the rooms and its radiant heat warms up the floors and furniture and renders zone control of the type described exceedingly valuable, especially in office buildings.

Some of the older very large hot water heating systems used mechanical circulation for the mains, but depended on thermal circulation for the risers, and had loop connections between these, with many radiators connected into the same riser; the supply connection to the radiator being some feet above the return connection. The pumpage of water necessary to secure results in such systems is enormous, and the temperature drop through any given radiator usually is exceedingly small. Re-arrangement of the piping so as to secure a positive differential is advisable with such old plants.

In general, the velocity of the water through the pumping unit in a new electrically circulated system on the basis of a 10 deg temperature drop should not exceed 2 fps in a plant using a single 2-in. main; 3 fps with the equivalent of a single 4-in. main; 4 fps with the equivalent of a single 6-in. main; 4.5 fps with a single 10-in. main; and 5 fps with a single 12-in. main or its equivalent. The velocities of water listed apply also to the mains in which the units are mounted. Reductions in pipe sizes of mains should be made as the successive radiation load is taken off, and the sizes of risers and radiator branches should be so selected as to maintain the same basic friction loss per foot of pipe throughout the system. Such velocities will work out to a temperature drop of around 10 F in the radiation in the coldest weather, providing that the circulating units are selected as recommended by the manufacturers.

It will be noted that on *old* systems, with all the piping in place and of larger pipe sizes than those suggested, a considerably smaller temperature drop than 10 F usually will be obtained. This will do no harm and will not prejudice the probability of savings.

One objection to hot water heating has been the fact that domestic or service water at adequate temperature has not been obtainable, and separate heaters for this service have been required. These usually burn a concentrated fuel and sometimes require an annual expenditure almost equal to that needed for heating the house.

This matter has recently been solved admirably. The solution requires mechanical circulation but easily may be applied to old thermally circulated systems. A large heat transfer unit is placed near the hot water boiler, receiving the water from the latter around its copper coils. The service water circulates within the coils. The boiler water is maintained at any desired temperature the year around—say 180 F. This easily heats the service water by thermal circulation to say, 160 F.

The hot water mains for the house heating are by-passed off the heater through a non-return valve, and the return from the house heating system passes to the mechanical circulator. The latter operates according to the dictation of a room located thermostat. In summer the mechanical circulator operates not at all, but a very small gas fire or oil fire or stoker fire or manually tended coal fire serves to keep hot the large volume of water in the well insulated boiler. In effect this boiler becomes an enormous thermos bottle.

In one instance out of many of which the authors have knowledge the following tabulation shows the remarkable savings which were effected by this adaption of service water heating. Column 1 gives the cost of heating with a gas burning equipment; Column 2 gives the cost of coal for heating the same demand of service water with the new stoker-fired arrangement:

Month	Column 1 (1932)	Column 2 (1933)
January	$23.00	$2.65
February	22.60	2.70
March	23.40	3.10
April	21.40	1.95
May	20.00	2.65
June	21.00	2.40
July	20.00	2.57
7 Months	$151.40	$18.02
(Saving in 7 months........$133.38)		

It is admitted, of course, that thus to increase arbitrarily at all times the temperature of the water in the boiler reduces the efficiency of heat transfer, fire to water. On the other hand small direct fired service water heaters in many cases are woefully inefficient. With oil as a fuel in the house heating boiler the service water problem sometimes has been bothersome. The savings to be made by the arrangement described are noticeable under such circumstances.

Some of the ultra-modern convector types of room-located heaters, often placed in a flue, have been found unsatisfactory with thermally circulated hot water. When mechanically circulated positive differential hot water is used these convectors become efficient. Many of the light weight convectors are so quick to heat and to cool when steam or vapor is used that they are con-

sidered unsatisfactory. The use of mechanically circulated water with its factor of stored heat often removes the objection.

Water boilers, whether under thermal or mechanical circulation, have cracked sections or burned crown sheets or explosions very rarely indeed. The water level in the system can vary a great many feet without danger of damage.

Old thermally circulated hot water heating systems easily have been changed to two-pipe steam systems. It is interesting to know that there are a number of two-pipe systems with the small return pipe sizes satisfactory for vacuum

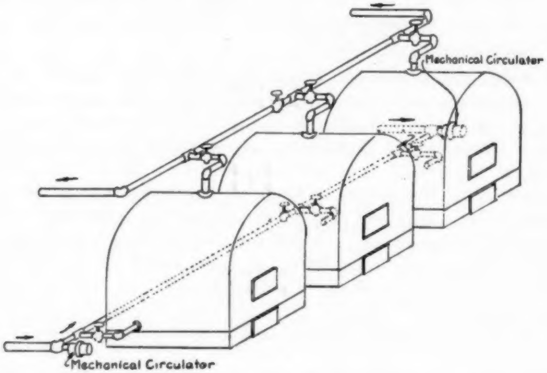


FIG. 3. TYPICAL MECHANICAL CIRCULATORS AS USED FOR LARGE GREENHOUSES

(The mains are often as large as 12-in. diameter)

pump return which have been changed to mechanically circulated water systems. There will be more of these. All that is necessary to do is to remove the traps from the radiators and the vacuum specialties from the boiler and to provide means for air venting and expansion. The pump may have an unusually high resistance to overcome with such small return piping, but usually will require less energy than did the vacuum pump or condensation pump which may have preceded it in service.

A mechanically circulated hot water system may be used as a summer cooling system if a supply of chilled water is available. Such a plant utilizes a maximum part of the investment the year around. The radiators used for cooling must have drain pans under them and should have fans under or near them to induce rapid air circulation.

The same radiators as are used for hot water heating will accomplish a very substantial amount of cooling under such circumstances. The insulation used on the pipes and surfaces of such a plant should be inert to the moisture which will condense out of the surrounding air. Such insulation, for heating and for cooling, is available in the market.

Conclusions

Heat transfer by forced liquid which is forced to circulate independently of thermal conditions has many advantages.

The value of having a variable mean effective temperature of the heat transmitting medium cannot be over-emphasized. Savings in fuel and in comfort invariably attend heating systems which adjust the mean effective temperature of the heating medium to suit the varying demand. Such systems reduce greatly the difficulty of temperature control.

They lend themselves admirably for adaption to cooling as well as to heating and are capable of providing service hot water as effectively as do steam heating plants.

Old liquid heating systems not having pumps can be improved and corrected by installing modern electric circulators.

DISCUSSION

Perry West: I should like to take issue with one of the statements or instructions in this paper, i. e. to design the pipes for the same loss per foot, because that would depend on whether or not the circuits were all the same length. The circuits should be designed so that each one would have the same total loss and all would be balanced. I think that is a mistake and I should like to call attention to it.

H. L. Alt: I can't answer Mr. West's question about the loss per foot because I don't know what these gentlemen have been figuring on; whether they mean that the friction loss per foot should be the same throughout the pipe system, as you would figure a forced circulation, or that the loss per foot should be such as to produce a total drop on the end of each circuit on a given amount. But the paper states that the loss per foot, which means that the shorter circuits would have a lesser drop than the longer circuits, might be overcome by using throttle valves on the return side of the circuit.

In sizing steam piping, we don't attempt to have the same drop in each branch. We have a short branch near the boiler and use the same basis of sizing it as we do in the long branch that runs a mile away from the boiler. I do believe The Guide makes the statement that, theoretically, we could use a higher drop on a short branch, but that is not generally done.

H. A. Snow[1]: If it is of interest, I am operating, in my own home, a vapor system with thermostatic traps on the radiators for hot water, and during the recent cold spell when the thermometer went to 17 below zero, I had no difficulty at all in heating my house with vapor radiation running hot water. That condition has been maintained not only in 17 below zero weather but also when the outside temperature was 50 F. I am doing this as an experiment and am simply offering the use of that apparatus if there is anything to be gained by accurate tests. I, personally, have made some interesting tests, having a thermometer on the supply and on the return, and have also taken some contact tests of temperatures of supply and return at the radiators. Those, of course, are not accurate tests but they are interesting, and if the Society is interested in that subject and cares to experiment with my particular system they are welcome to do it.

[1] Buerkel and Co.

B. F. BURT: The sizes which I recommended were general only. The scope of the paper would not permit going into much detail on this point.

For example, consider a system that will require a total of 350 gpm with a total equivalent length of longest circuit of 300 ft, using also a shorter circuit of say 250 ft. A certain size of pump will be good for 350 gpm through an equivalent circuit of 310 ft of 6 in. pipe. Of course, only a few feet of the circuit will be 6 in. pipe. A 6 in. circulator will move 350 gpm at a delivery head of approximately 26 in. of water, and the head per equivalent foot of pipe for the longest circuit will therefore be 26 divided by 350, or .0743 in. Every section of this longest circuit, and all branches from this longest circuit, may be so sized that the drop per foot will be approximately this figure, although in some cases it might be well to figure the branches separately in a direct return job.

Every different section of piping is figured so that its friction per *equivalent* foot will be not materially more than the value set for this particular main. In the case of a materially shorter main, and this seems to be the question which Mr. West brings up, we recommend that a new available head per foot of circuit be used, and since the head developed by the circulator will be the same as for the longer circuit, it necessarily follows that the head *per equivalent foot* of pipe for the shorter circuit will be greater than for the longer main, and therefore smaller pipe sizes will be satisfactory for many of the sections of this shorter main, its risers, etc.

Of course, since pipes are only available in commercial sizes, exact sizing is not possible unless some system of orifices (or the like) be used, and we do not consider this worth the trouble, if the system is designed to have a small enough maximum temperature drop in the first place, for then a glaring difference in water flow cannot cause much difference in mean effective temperatures between *any* two radiators. This is one of the valuable characteristics of electrically circulated hot water.

No. 972

DESIGN AND EQUIPMENT OF THE PIERCE LABORATORY *

By C.-E. A. WINSLOW,** LEONARD GREENBURG † (*MEMBERS*), L. P. HERRINGTON † (*NON-MEMBER*), AND H. G. ULLMAN ‡ (*MEMBER*)
NEW HAVEN, CONN.

GENERAL OBJECTIVES

THE John B. Pierce Foundation was established under the will of the late John B. Pierce and as specified in the will, the object of this Foundation "shall be for the promotion of research, educational, technical or scientific work in the general field of heating, ventilation and sanitation for the increase of knowledge to the end that the general hygiene and comfort of human beings and their habitations may be advanced." The Foundation officers are Clarence M. Woolley, president; Rolland J. Hamilton, vice-president, and Henry L. Weimer, secretary and treasurer. For nearly two years work has been in progress on the design and construction of a laboratory for researches relating to the physiological effects of various atmospheric conditions, particularly those due to radiation of various types and electrically charged particles. The Laboratory of Hygiene was formally opened on November 4, 1933. While an independent institution, it will be operated in affiliation with the Department of Public Health of the Yale School of Medicine, under the direction of the senior author.

Since the approach must be a broad one, primarily from the standpoint of human health and comfort, the staff has been so organized that medicine, physiology, and psychology, as well as engineering and physics, are represented by competent experts.

The building (Fig. 1) is situated at 290 Congress Ave., New Haven, in immediate proximity to the Yale Medical School. It is a two-story brick structure, with basement, occupying a total ground area of approximately 1,800 sq ft. In general, the basement is occupied by air-conditioning machinery and animal rooms; the first floor by air-conditioning machinery, workshop and offices; and the second floor by test rooms and a laboratory.

* Contribution No. 1, John B. Pierce Laboratory of Hygiene.
** Director, John B. Pierce Laboratory of Hygiene.
† Associate Directors, John B. Pierce Laboratory of Hygiene.
‡ Consultant.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by C.-E. A. Winslow.

TEST HOUSE

The heart of the installation is a two-room frame house completely surrounded, above and below and on all four sides, by shell spaces. In these shell spaces any desired conditions of temperature and humidity can be produced. The location of shell spaces and experimental rooms is indicated in Fig. 2 (the second floor plan) and in Fig. 3.

The plant is, of course, designed somewhat along the lines of the laboratories of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at the United States Bureau of Mines, Pittsburgh;[1] of the American Radiator Co. at Yonkers,[2] and of the University of Illinois.[3] The chief differences are, that the air-conditioning of the shell space is more elaborate at New Haven than

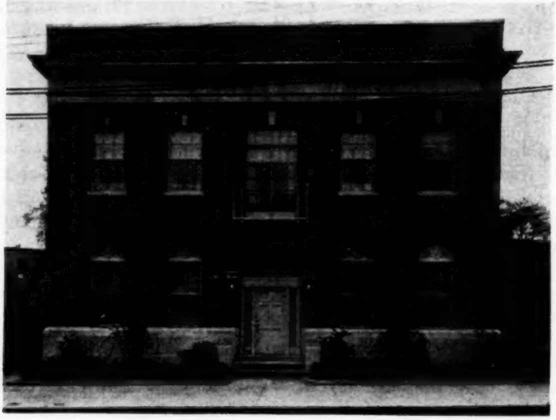


FIG. 1. THE PIERCE LABORATORY

at Urbana or Yonkers; that the test rooms are of standard size and ordinary house construction, furnished in such a way as to promote normal psychological conditions in the subjects studied, and forming a suite which can be used for long-time tests on subjects in continuous residence for weeks; and that the facilities for automatic records of both physical and physiological reactions are unusually complete.

The interior building containing the test rooms in which studies are to be actually conducted is of standard frame construction. It is 29 ft wide and 16 ft

[1] A Study in Heat Transmission with Special Reference to Building Materials, by F. C. Houghten, A. S. H. V. E. TRANSACTIONS, Vol. 28, 1922, p. 81.
Some Physiological Reactions to High Temperature and Humidities, by W. J. McConnell and F. C. Houghten, A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923, p. 129.
Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yagloglou, A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923, p. 163.

[2] The Heating Effect of Radiators, by C. W. Brabbée, A. S. H. V. E. TRANSACTIONS, Vol. 32, 1926, p. 11.

[3] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 77.

deep and includes two rooms, each 15 ft by 12 ft by 9 ft high,[4] separated by a corridor 4 ft wide. This corridor includes first an entrance lobby about 4 ft square. As one enters this lobby through a door from the front shell space, doors open on either side into the two test rooms. A fourth door opposite

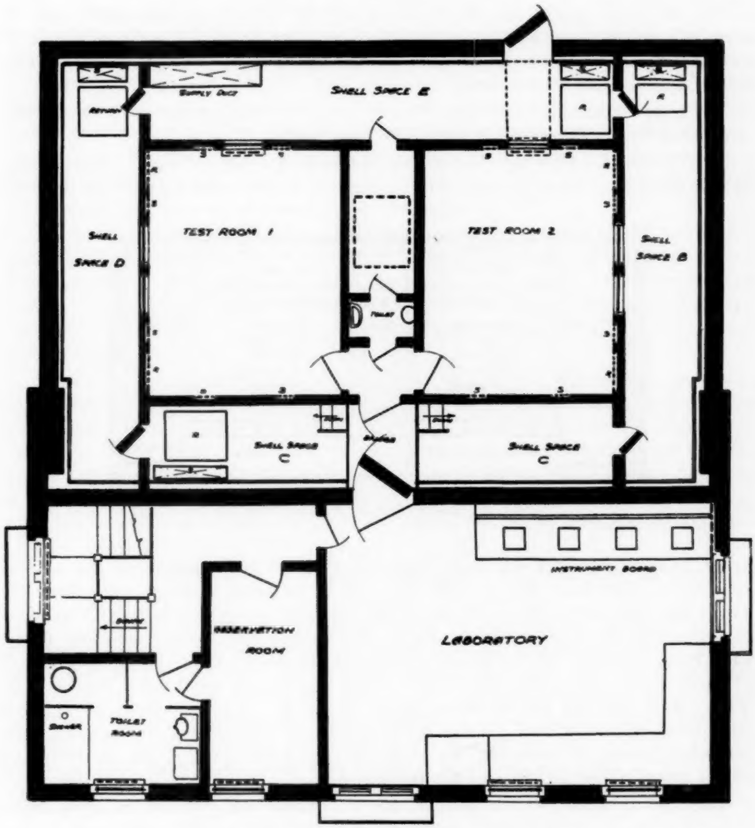


FIG. 2. PLAN OF SECOND FLOOR

opens into a toilet room, 4 ft by 3 ft, behind which is a storage room 8 ft by 4 ft. The lobby and toilet room can thus be combined with either test room or with both to form a suite for experiments in which subjects are to be kept under specified conditions for days or weeks. The walls separating the corridor from the test room are of wood lath and plaster. Above test rooms and corridor

[4] It will be noted that the test rooms are of the standard dimensions prescribed in the Code for Testing Radiators, A. S. H. V. E. TRANSACTIONS, Vol. 33, 1927, p. 19.

is an attic space 23 in. high with a flat roof of boards covered with tar paper, separating it from the upper shell space.

CONTROL OF SHELL SPACES

The six shell spaces are all insulated on the outside by laminated cork of standard construction and are separated from each other by walls of Portland cement and cork. Entrance to all the shell spaces is through standard refrigerator doors, 6 in. in thickness.

The temperature and humidity in each of the six shell spaces is independently controlled by conditioned air, supplied as follows:

Shell space *A* (over the roof of the test house) is 35 in. high. It is divided into three areas, by walls of Portland cement, plaster and cork, running from

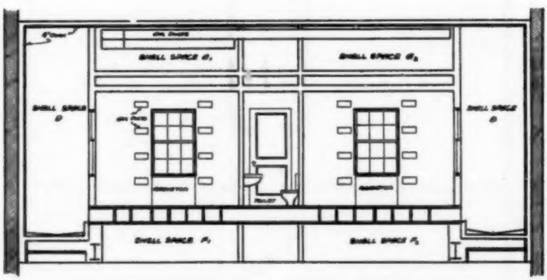


FIG. 3. LOCATION OF SHELL SPACES

front to back over the partitions between the two test rooms and the central corridor. (See Fig. 3.) Conditioned air is introduced either into space A_1 (over the left room) or space A_2 (over the right room) or into both at once, through a single duct 12 in. x 36 in. which branches to two separate ducts, each 8 in. x 36 in., supplying either half of the upper shell space. Air leaves A_1 and A_2 by the space above the corridor which is used as a return duct. Thus, it is possible to maintain either space A_1 or A_2 or both at a given condition. If conditioned air is supplied to one of these upper shell spaces only, the other cannot be controlled but is left to adjust itself to a value between that of the room below and that of the outdoor air.

The shell space under the floor of the test rooms and corridor is 25 in. high. Otherwise it is divided and controlled in a manner exactly analogous to the ceiling shell space. Shell space F_1 is under test room 1 at the left as one enters; shell space F_2 under test room 2 at the right.

The four lateral shell spaces, *B* at the right-hand wall of test room 2, *C* in front of the entire test house, *D* at the left of test room 1, and *E* in the rear of the entire test house, are all between 51 and 55 in. wide and all independently controlled by conditioned air. (See Fig. 2.) The air enters from a ceiling supply duct extending the entire length of each shell space and perforated throughout its length with 1-inch holes. Air leaves each shell space through a floor grille 3 ft square located at one end.

The exhaust ducts from shell spaces *B, C, D* and *E* lead to four separate conditioning chambers. *A* and *F* are on the same main supply and exhaust ducts connecting with a fifth conditioning chamber. Thus, either the ceiling or the floor shell space or both, can be maintained at a given temperature and humidity. If one only is controlled the other may be shut off and allowed to adjust itself to the temperature of adjacent areas.

The five conditioning chambers are all alike, and each includes a dry-coil refrigerating unit of ample capacity; and a spray chamber for humidification. The refrigerating unit is a standard ammonia compressor. The spray chamber has four spray nozzles provided with recirculated water by a centrifugal pump and a small steam heating unit for raising the temperature of the water.

After leaving the conditioning chamber the air passes to the vertical supply duct in which is located the air heater. This is a fin-type construction with approximately 80 sq ft of steam-radiation capacity.

Dry-bulb temperature is controlled by steam valves governed by thermostats located in the return air ducts. As heat is called for in the shell spaces, one or both of two steam valves connected with the air heater will automatically open. As refrigeration is required, these steam valves automatically close and the ammonia compressor is started at a speed corresponding to the load requirements.

Humidity is controlled by humidostats located in the return air ducts, and governing the temperature of the spray water and the proportion of recirculated air which passes through the spray chamber. When an increase of humidity is desired a larger proportion of air is passed through the spray chamber and the steam supply to the water heater is increased. When a decrease is called for, a larger proportion of the recirculating air is by-passed around the spray chamber and the remainder of the air is dehumidified by passing through the refrigerating unit.

The five conditioning units for the shall spaces are located in the basement and are shown in Fig. 6.

The designers of this equipment guarantee in any part of the shell space a minimum average dry-bulb temperature of 0 F, plus or minus 0.5 F, with a maximum summer outside atmospheric temperature not over 90 F dry-bulb and 75 F wet bulb and with inside test room temperature not over 85 F. They also guarantee dew-point temperatures from 40 to 95 F, the maximum variation in wet-bulb temperature at the point of control to be not over 0.5 plus or minus. This latter guarantee covers the dry-bulb range from 50 to 100 F, and will hold regardless of outside temperature and humidity conditions.

THE TEST ROOMS

The test rooms themselves, as pointed out before, are two in number, each 15 ft by 12 ft by 9 ft high. Each room has two windows opening on the end shell space and one window opening on the rear shell space. Outdoor scenes are painted on drops hung on the wall of the shell space outside each window. Each room is provided with the following facilities for heating and ventilation, which can be used separately or in combination:

(1) *Direct Steam Heating.* Each room is provided with two steam radiators placed under the windows. These radiators are of cast iron column type and can be

set either as free standing or concealed radiators. Provisions have also been made for using a hot-water system of heating if desired.

(2) *Radiation.* Special electrical outlets are provided for the operation of radiant panel heaters and certain special heaters of this type have been designed for our experimental purposes. In addition, it is possible by heating a given shell space to a high temperature to use ceiling, floor or one wall as a low-temperature radiant-heating unit.

(3) *Indirect Heating.* Conditioned air may be supplied to each room by any or all of six different supply ducts. Two of these ducts run up the front, rear and end wall of each room. (Fig. 2.) Each duct opens into the room by five wall registers (each 9 in. x 13 in.) at different levels. (Fig. 3.) The center of the upper register is 12 in. below the ceiling and that of the lowest, 6 in. from the floor. It is possible, by manipulation of dampers, to introduce air at any desired level on one or all of three sides of the room in order to study physiological reactions to temperatures at different levels.

Two outlet grilles (6 in. x 27 in.) are located on the side wall at the floor of each room. They connect by means of return air ducts with the test room con-

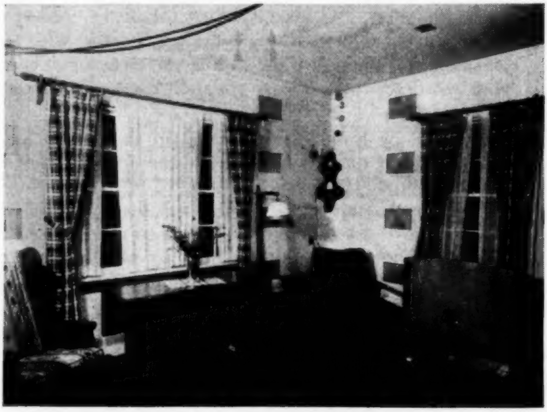


FIG. 4. CORNER OF TEST ROOM

ditioning-plant on the first floor. This is a commercial type forced-air heating unit of small size with the addition of refrigerating coils and a water jet for humidification. It was thought best to use simple commercial-type apparatus in this case in order to make the tests as practical as possible. This part of the plant will be more fully discussed in a later paragraph. If more exact regulation of test-room humidities than this equipment permits should be desired, cross-connections can be made with one of the shell space units.

(4) *Gravity Exhaust.* In the ceiling of each test room is a register (20 in. square) connecting with a gravity-exhaust duct passing up through shell space *A* and the roof of the laboratory building to the outside.

(5) *Gravity Supply of Outdoor Air.* In test room 2 (but not in test room 1) provision is made for a supply of fresh and untreated outside air for comparison with various forms of conditioned air. Opposite the lower half of the rear window is an opening about 30 in. square through the outer wall of the rear shell space to the outer air. (See Fig. 2.) Ordinarily this opening is closed by a refrigerator door. When untreated outdoor air is desired, an insulated section of 30-in. duct can be let down from the ceiling of the shell space to connect the opening in the outer wall of the shell space with the lower half of the window in the test room. With

this section of duct in place and with the outer refrigerator door and the lower half of the window open, outdoor air may enter by gravity, passing only through about 5 ft of a 30-in. square duct which should not materially alter its properties.

The test rooms are lighted electrically by four wall brackets and four standing lamps and are furnished as attractively as possible with rugs and hangings, and household furniture. Each room contains a couch-bed (which can be supplemented by cots if long-continued residence of several persons is desired), a desk, a large table and comfortable chairs. Fig. 4 shows a corner of one of the rooms.

Apparatus for Control and Automatic Records

In the laboratory on the second floor adjacent to the test rooms (see Fig. 2) is a large panel board. One-half of this panel board is occupied by remote-control switches which govern the air-conditioning equipment for the shell spaces. One set of five switches controls dry-bulb temperatures, another set of five switches, the wet-bulb temperatures. The conditions in the return duct from each shell space are recorded in the form of dry-bulb and wet-bulb temperatures registered continuously on two multiple-point recorders, one for the dry-bulb and one for the wet-bulb temperatures.

Air conditions in the test rooms are governed by thermostats and humidostats placed on the wall near the door of each room. Very complete provision is made for recording dry-bulb temperatures (of atmosphere, walls or subjects as desired) and of wet-bulb temperatures of the air of the test rooms. A terminal board concealed behind a hinged bookcase on one wall of each room contains connections for 51 pairs of wires which will terminate in thermocouples to be placed at the sites whose temperatures are to be recorded. From the terminal board the wires pass through the wall to the panel board in the laboratory. Here 32 pairs of wires from each room are connected to bus-bars from which these paired wires lead to 16-point continuous recorders. The other 19 pairs of wires from each room lead directly to a switch box wired in series with a potentiometer so that the E. M. F. may be read directly. In addition, the bus-bars previously mentioned are so wired that the E. M. F. utilized in the automatic recorders may be switched over to the hand-switching set and read on the potentiometer. Some of the thermo-junctions will be equipped with wicks and water feed devices to record humidity.

Outlets are provided in each test room for six resistance thermometers which when connected give continuous records of air temperature in the test rooms. They register on two additional recorders on the panel board table in the laboratory.

Special Laboratory Equipment

The laboratory on the second floor (Fig. 2) has a floor space of approximately 400 sq ft. Besides the control panel board and recording apparatus, it includes laboratory tables, a fume hood, an instrument cabinet and a drawing table. It is equipped with gas, electricity, vacuum and compressed air, and a steam bath.

At one side of the laboratory is set an ionizing apparatus of the type designed by Prof. F. Dessauer of Frankfort, Germany, which is so placed as to deliver ionized air by means of a glass tube passing through the wall into the small observation room adjacent. (Fig. 2.) In this room, two subjects at a time,

reclining on couches, may breathe the ionized air. A toilet and shower room occupy the remaining corner of this floor.

If it is desired to supply ionized air to the test rooms, this Dessauer apparatus which produces large ions, or a special apparatus yielding small ions, can be

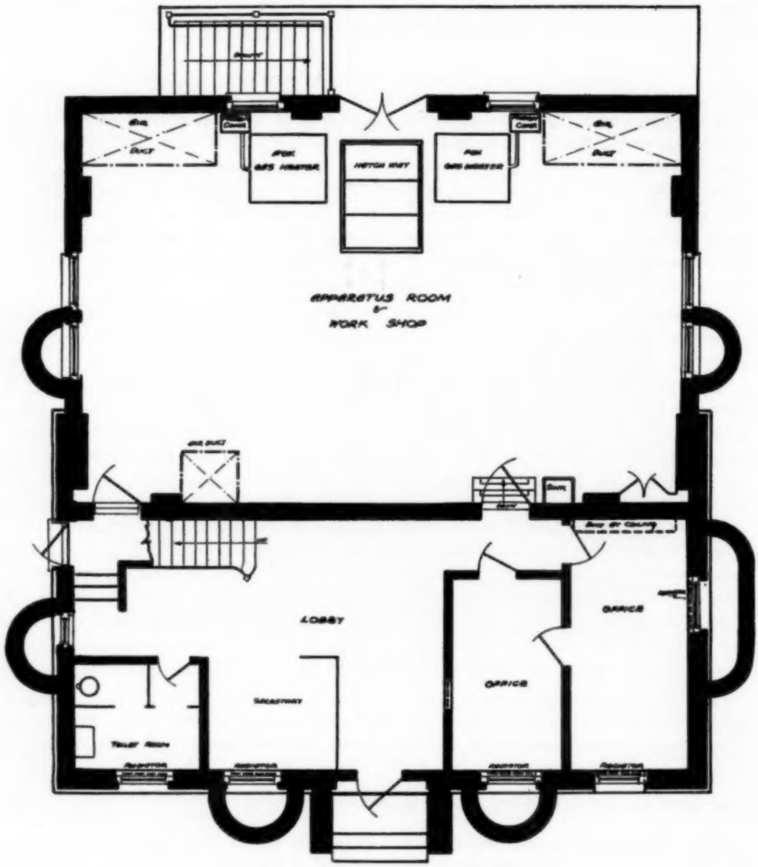


FIG. 5. PLAN OF FIRST FLOOR

moved to the test rooms and the ions introduced directly into the room or into the air supplied to the room through its inlet ducts.

The laboratory equipment includes a portable Dessauer ion-counter for determining the ion content of indoor or outdoor air; a Benedict-Roth basal metabolism machine, kymographs and other usual physiological apparatus.

FIRST FLOOR

The first floor of the building includes a reception room with a partitioned office for a secretary, two offices for the director and associate directors, and a large room at the rear. (See Fig. 5.) The latter is occupied in part by the

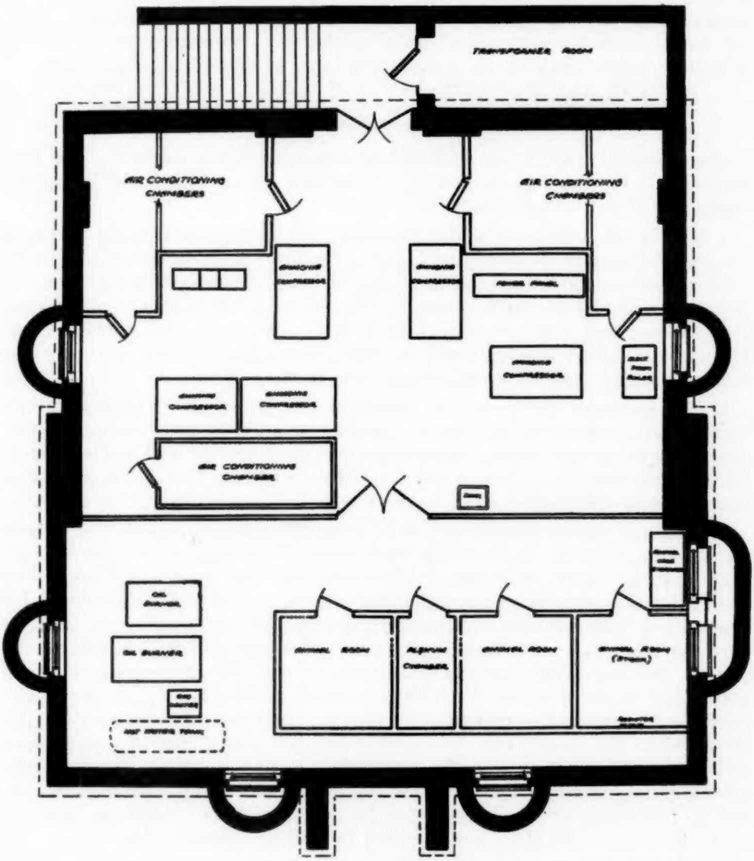


FIG. 6. PLAN OF BASEMENT

air-conditioning apparatus for the two test rooms while the rest serves as a workshop. The workshop is provided with a small machine lathe, a drill-press, a combination circular saw and planer, a band saw and a grinder.

The conditioning equipment for each of the two test rooms includes, as pointed out above, a forced-circulation gas-heating unit in combination with a

refrigerating and humidifying unit. Each conditioning chamber is equipped with a variable-speed fan for circulating the air to and from the rooms, and dampers provide for any desired mixture of outside and recirculated air. The refrigerating units are compressors employing dichlorodifluoromethane as refrigerant. Their capacity is such that both of the test rooms can be cooled to 55 F, when the surrounding shell space is at 90 F. With both compressors connected to one test room, a temperature of 40 F can be maintained in this one room under the above shell-space conditions. Humidification is effected by a small single spray of water controlled by an electrically operated valve.

THE BASEMENT

The basement floor is occupied by the equipment for conditioning the shell spaces, and for heating the laboratory building as a whole, and by special rooms for animal experimentation.

A large room in the rear is devoted chiefly to the five conditioning chambers for the shell spaces as shown in Fig. 6. Each of the five units includes the mixing and refrigerating chamber for its respective shell space, and the ammonia compressor which serves it. In the same room is a small electric high-pressure boiler for supplying steam to the laboratory and for sterilizing animal cages. Here also is located the main electric panel board for controlling light and power supplied to all parts of the building.

In the northeast corner of the basement are placed three boilers; an oil burning boiler supplying heat for the building as a whole; a second oil burning boiler providing heat to the conditioning chambers for the shell spaces and to the steam radiators in the test rooms; and a small gas-fired boiler supplying heat to the animal rooms.

The animal rooms themselves are in a special suite constructed of hollow tile. At the west end is an animal stock room, 7½ ft by 8 ft by 10½ ft high. This room is heated by direct radiation and has one outside window. The rest of the suite is occupied by two animal test rooms (each the same size as the stock room) with a smaller air-conditioning room between them. The walls of all these rooms are of hollow tile and the floors of concrete drained to the center. The two animal test rooms are supplied with air which enters near the ceiling and leaves near the floor, by a fan and heating unit in the small air-conditioning room. This unit provides only for tempering the outside air. The air in animal test room 1 can, however, be supplied with completely conditioned air by a cross-connection with the supply ducts to shell space *C*. Special lamps provide ultra-violet radiation since the two animal test rooms have no windows. Special provisions are being made for the introduction of ionized air into one of the test rooms and plugs are provided for radiant panel heating.

It is hoped with this plant as a whole to be able to investigate the reactions of both human beings and animals to a wide variety of atmospheric conditions, particularly those associated with the subtler physical variables, and to study the operation of some of the newer procedures in the field of heating and ventilation in rooms of ordinary dwelling-house construction exposed to fully controlled external weather conditions.

No. 973

SELECTING TEMPERATURES AND WIND VELOCITIES FOR CALCULATING HEAT LOSSES

By PAUL D. CLOSE* (*MEMBER*), CHICAGO, ILL.

PROBABLY the most important of all heating calculations is that of estimating heat losses. Regardless of how large or how small a building may be, some estimate of the heat to be supplied should be made if a modern heating system is to be installed. While it is a comparatively simple matter to design a heating plant of *ample* capacity to maintain the desired conditions, the difficult problem is to provide just the right amount of heating capacity, and no more, since any excess means an unnecessary increase in the cost of equipment.

The process of determining the size of heating plant required involves the equating of two quantities, namely (1) the so-called heat loss of the building or the probable amount of heat which will be dissipated from the structure for certain specified inside conditions and assumed outside conditions and (2) the heat gain or the heat to be supplied by the heating apparatus. Often, however, a factor of safety, or more properly, a "factor of ignorance," is used in the calculations to allow for quantities of unknown or doubtful magnitude, such as uncertainties of the weather, inadequate data concerning the rate of heat loss of the building structure, and doubt as to accuracy of the assumptions under service conditions.

FACTORS GOVERNING HEAT LOSSES

There are many factors which influence or have a bearing on the loss of heat from a building. These may be broadly classified as (1) inside conditions, (2) outside conditions and (3) the building. Under *inside conditions* may be included the temperature to be maintained at a stated level, stratification of heated air due to high ceilings, type of heating system and manner of distribution, period and nature of occupancy, method of temperature regulation (whether manual or automatic), and the influence of any outside air that may be introduced for ventilating or air conditioning purposes.

Outside or *weather conditions* include temperature, direction and velocity of wind, humidity, precipitation, and percentage of sunshine; although the latter probably may be neglected in heating calculations as it is logical to assume

* Chief Engr., Industrial Uses Div., Celotex Co.
Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

that the period of maximum heat loss occurs when there is no sunshine. The *building* includes the size, shape, the type and quality of construction, and the physical properties of the materials used, especially the rate of heat transmission and the heat capacity thereof.

METHODS OF CALCULATION

Briefly the problem is to determine the maximum heat loss from the building for some interval of time giving consideration to the coordinated effect of all of the foregoing factors.

The first formulas for calculating heat losses such as the Carpenter, the Mills and the 4, 4, 4 rules took account only of the volume of the room or space under consideration and the area of wall and glass surfaces, disregarding the type and quality of construction and the materials used. It made no difference

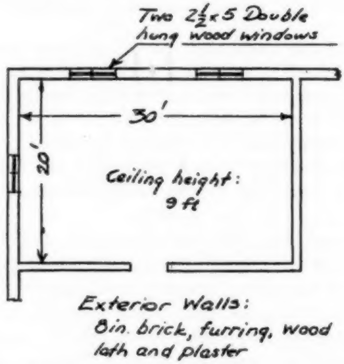


FIG. 1. DETAILS OF ROOM NO. 1

how well or how poorly insulated a building might be, or whether storm sash or weatherstrips were used, the same size or capacity of heating unit would be required. These rules were "satisfactory" in most cases only because they were safe. They are still used to some extent, but are not recommended except for checking purposes.

The Btu method outlined in Chapters 2, 3 and 4 of THE A. S. H. V. E. GUIDE 1933, is the most accurate method of calculating heat losses at present known. In this method the heat losses are divided into two kinds namely (1) transmission losses and (2) infiltration (or exfiltration) losses. Transmission losses are those resulting from the transfer of heat by conduction through solid members of a building when a difference in temperature exists between the two sides. Infiltration losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building shell such as cracks around doors and windows, fireplaces, and chimneys.

In making the calculations, certain temperatures are assumed. The inside temperature to be maintained at a given level in the room is specified. The outside temperature is assumed to be a certain number of degrees—usually 10 or 15

—above the lowest temperature on record for the district in which the building is located. Special allowances are made for temperatures in unheated spaces.

The wind velocity selected for making the infiltration calculations is usually the average during the months of December, January and February. Less importance is assigned to the effect of wind velocity upon the transmission losses and usually a velocity of 15 mph is selected for all except thin, highly-conductive walls. The only effect of variations in the wind velocity upon the transmission losses is to change the heat resistance of the exterior surface; and since this resistance is ordinarily only a fraction of the total heat resistance of the wall (except for highly-conductive constructions), it is sufficiently accurate to assume a single value for practically all surfaces, as already

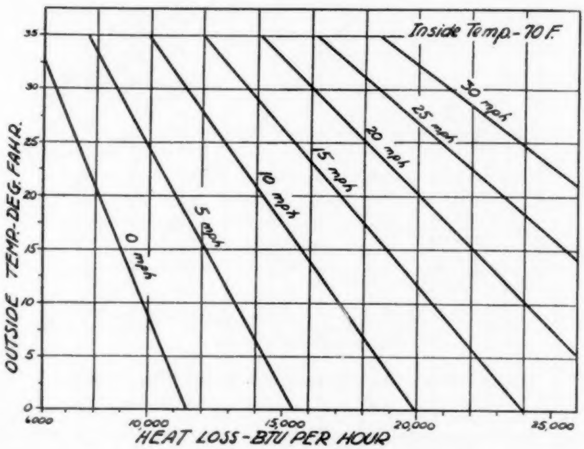


FIG. 2. HEAT LOSS CURVES FOR ROOM NO. 1[a]

[a] *Note:* The heat loss curves do not show a constant or consistently varying rate because the infiltration factors upon which the calculations were based do not change the same amount for equal increments of wind velocity. (See factors for poorly-fitted non-weatherstripped double-hung wood windows, Table 2, p. 44, A. S. H. V. E. GUIDE 1933.) Attention is also called to the fact that the transmission losses were corrected for varying wind velocities and that these losses increase as the wind velocity increases. In the case of the transmission losses through glass, this is an important factor.

stated. After the transmission and infiltration losses are computed for all rooms or spaces to be heated, an additional allowance is made for rooms exposed to prevailing winds.

DISCREPANCIES BETWEEN ACTUAL AND COMPUTED RESULTS

Although heat losses calculated according to the foregoing methods have been found to be reasonably accurate in the majority of cases, occasionally discrepancies arise between the actual and the computed results, particularly where unusual climatic conditions are involved and the diffusivity [1] of the construction

[1] The *diffusivity* is a constant which takes into account the resistance to heat flow and the heat capacity of the structure and is expressed by the following formula:

$$h = \sqrt{\frac{k}{cp}}$$

where

k = conductivity
c = specific heat
p = density

is low. It is presumed therefore that such discrepancies may be attributed largely to (1) improper selection of temperatures and wind velocities, and (2) neglect of or improper allowance for the diffusivity of the structure. It is the purpose of this paper to discuss the selection of temperatures and wind velocities.

As previously stated, the usual procedure is to select an outside temperature above the lowest recorded for a period of years in the given locality. On the other hand, the wind velocity used in calculating the infiltration losses is the average during the months of December, January and February. This wind

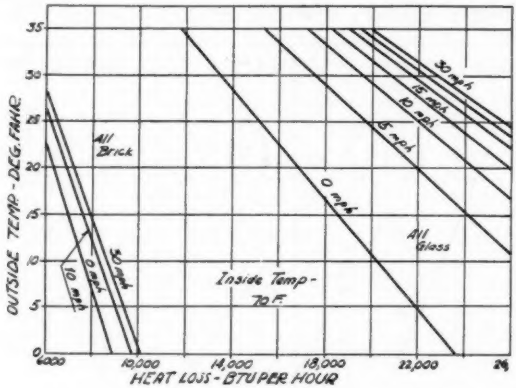


FIG. 3. HEAT LOSS CURVES FOR ROOMS NOS. 2 AND 3

velocity and the temperature selected are applied together in arriving at the infiltration losses and yet they may have no relationship to each other. Since the problem is to determine the worst combination of weather conditions, the two most important of which apparently are the temperature and wind velocity, it seems logical that the temperature and wind velocity selected should be those *occurring at the same time*, or the averages *during some interval of time*.

ANALYTICAL STUDIES OF TYPICAL CASES

In order to test the accuracy of the foregoing statement, an analysis was made of the heat losses from various types of construction for various temperatures and wind velocities, and the results compared with actual meteorological records for the city of New York. The diffusivity was neglected. The types of construction analyzed are as follows:

1. A room having two exposures and with walls of 8-in. brick, wood lath and plaster. Details of the construction are shown in Fig. 1. The windows were assumed to be poorly fitted and non-weatherstripped so that the infiltration losses would be high.

2. A room of the same dimensions and similar to (1) but with no windows.

This type was selected to show the effect of variations in temperature and wind velocity when there are no infiltration losses.

3. An all-glass room of the same dimensions. This type was selected to show the effect of changes in temperature and wind velocity on constructions of high rates of transmission, assuming no infiltration. Solar radiation and reradiation were neglected.

The surface coefficients used in the calculations for various wind velocities were derived from an A. S. H. V. E. research paper entitled, Surface Conductances as Affected by Air Velocity, Temperature and Character of Surfaces, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.) Infiltration data and other coefficients were taken from THE A. S. H. V. E. GUIDE, 1933. The method of calculating heat losses was that outlined in Chapter 2 of THE GUIDE, 1933, except that no exposure factors were used.

TABLE 1. TEMPERATURES AND WIND VELOCITIES FOR NEW YORK CITY FOR DAYS ON WHICH THE MEAN TEMPERATURE WAS + 14 F OR BELOW[b]

(January, 1921, to February, 1933, inclusive)

Date	Temperature			Wind		Per cent of Possible Sunshine
	Max.	Min.	Mean	Average Velocity	Prevailing Direction	
1-18-21	16	6	11.0	16.8	N W	100
1-19-21	22	4	13.0	6.2	N N W	100
1-25-21	20	4	12.0	20.3	N N W	100
1- 2-22	16	7	11.5	19.8	....	100
2-17-22	17	− 2	7.5	10.8	....	100
12-30-23	22	7	14.5	9.6	N W, N N W	100
1-27-24	18	5	11.5	13.8	W N W	100
1-23-25	27	2	11.5	21.4	N N W	100
1-24-25	24	2	13.0	10.5	W S W	91
1-28-25	14	− 2	6.0	15.4	N N E	93
12-27-25	16	8	12.0	19.8	N W	100
1-29-26	23	5	14.0	15.9	N W	100
2-11-26	21	8	14.5	17.7	N N W	100
12-18-26	20	8	14.0	17.0	N N W	100
1-16-27	22	7	14.5	18.1	N N W	100
1-27-27	25	− 1	12.0	6.9	N	94
2-16-30	30	7	18.5	14.2	N N W	100

[b] Data from New York Meteorological Observatory.

Fig. 2 shows the relation between heat loss, temperature and wind velocity for Room No. 1 shown in Fig. 1. Fig. 3 shows similar curves for Rooms Nos. 2 and 3.

The curve for a wind velocity of 0 mph (Fig. 2) represents *no infiltration.* In other words, for still air conditions, the heat loss of the building obviously is due entirely to the transmission of heat through the brick and glass on account of the difference in temperature on the two sides. As the wind velocity increases the heat loss increases rapidly mainly as the result of infiltration. For this particular case, the heat loss is doubled *for the same outside temperature* if the wind movement changes from 0 mph to about 13.3 mph, and is

tripled *for the same outside temperature* if the wind movement changes from 0 mph to 26.7 mph.

On the other hand, it is apparent from Fig. 3 that wind velocity has relatively little effect upon the heat loss from the solid brick wall construction without windows (Room No. 2), since there is no infiltration loss, the only difference being that the exterior surface resistance of the wall decreases as the wind velocity increases. The curves at the left indicate that a change in the wind velocity from 0 to 30 mph increases the heat loss only 13.5 per cent, as compared with 227 per cent in the case of Room No. 1 for the same increase in wind velocity. An intermediate result is obtained in the case of Room No. 3 consisting of all-glass walls, which are assumed to be impervious to air leakage. The increase in heat loss for the same outside temperature is 69.5 per cent, which is due entirely to the decrease in the exterior surface resistance of the glass.

From the foregoing analysis, the following corollary may be deduced:

A. Transmission losses depend upon the temperature head, and to a minor extent (except in the case of highly-conductive walls), upon the wind velocity; infiltration losses depend upon both the temperature head and the wind velocity. There is no general correlation between transmission and infiltration losses nor between temperature and wind velocities for all types of construction and climatic conditions.

One common method of calculating heat losses is based on the assumption that one mile of wind velocity is equivalent to one degree drop in temperature.[2] If the foregoing corollary is true, the accuracy of this assumption is questionable. Fig. 2 indicates that the heat loss for Room No. 1 will be about 21,200 Btu per hour when the outside temperature is 30 F and the wind velocity is 30 mph. If the theory were correct, these conditions would be equivalent to 0 deg F and 0 mph. However, the heat loss for these conditions will be only 11,500 Btu per hour for Room No. 1 according to Fig. 2, or less than one-half the actual heat loss. In the other two constructions (see Fig. 3), the differences resulting from the use of this rule are smaller, but the case cited, although perhaps extreme, illustrates the danger of applying an empirical rule of this nature to all conditions.

APPLICATION OF DATA TO NEW YORK CITY CONDITIONS

Climatological data for New York City from January, 1921, to February, 1933, are given in Tables 1 and 2. Table 1 is for cold days (mean temperature + 14 F or below) and Table 2 is for windy days (average wind velocity

[2] The actual drops in temperature per mile of wind velocity for various temperature ranges obtained in certain tests were:

Temperature Range (Deg. Fahr.)	Temperature Drop per Mile of Wind Velocity (Deg. Fahr.)
50 F to 40 F	0.75
40 F to 30 F	1.0
30 F to 20 F	1.1
20 F to 10 F	1.2
10 F to 0 F	1.3
0 F to 10 F	1.4
−10 F to −20 F	1.5

Note: See paper entitled, Establishment of a Standard Coefficient for Heat Losses Affected by Wind Movement, by H. W. Whitten and R. C. March, A. S. H. V. E. TRANSACTIONS, Vol. 23, 1916.

22 mph or higher). Significant so far as calculating heat losses is concerned is the fact that the coldest days were almost invariably clear, there being only three days out of seventeen when there was less than 100 per cent of the maximum possible sunshine, these three days averaging approximately 93 per cent. On the other hand, the windy days were almost invariably cloudy, the average percentage of maximum possible sunshine being 32.1 per cent, there being only two 100 per cent sunshine days out of 24. The probable explanation of the cloudiness on windy days is that a windy day usually signifies the approach or the existence of a storm.

TABLE 2. TEMPERATURES AND WIND VELOCITIES FOR NEW YORK CITY FOR DAYS IN THE MONTHS OF DECEMBER, JANUARY AND FEBRUARY WHEN THE AVERAGE WIND VELOCITY WAS 22 MPH OR HIGHER[b]

(January, 1921, to February, 1933, inclusive)

Date	Temperatures			Wind		Average of Possible Sunshine
	Max.	Min.	Mean	Average Velocity	Prevailing Direction	
1-17-21	38	16	27.0	23.1	N W	100
2-20-21	35	22	28.5	24.5	N E	0
1-11-22	36	26	31.0	26.3	N E	0
1-28-22	32	25	28.5	26.6	N E	0
1-29-22	35	27	31.0	24.2	N N E	38
12-28-22	39	21	30.0	32.0	E N E	0
1- 7-23	24	21	22.5	22.1	N	0
2-15-23	20	12	16.0	22.2	W N W	92
1-26-24	20	7	13.5	24.3	W N W	92
1- 2-25	28	22	25.0	27.0	E N E	0
1-12-25	29	21	25.0	23.0	E N E	0
12- 2-25	44	39	41.5	25.3	E N E	1
12- 3-25	49	39	44.0	30.6	N N E	0
12- 4-25	51	46	48.5	24.3	E N E	0
1-22-26	40	17	28.5	22.3	N N W	66
1-28-26	36	7	21.5	25.0	W N W	81
2- 4-26	30	25	27.5	24.0	N N W	14
12- 8-26	40	30	35.0	31.1	W S W	13
2-19-27	40	25	32.5	23.5	N E	10
2-20-27	25	23	24.0	30.8	E N E	0
12- 4-27	32	23	27.5	28.4	E N E	0
1-25-28	54	26	40.0	24.9	W N W	86
1-29-33	35	30	32.5	24.5	N N W	78
2-27-33	38	24	31.0	22.2	N N W	100

[b] From New York Meteorological Observatory.

The mean temperatures and average wind velocities indicated in Tables 1 and 2 are plotted in Fig. 4. The line *TWV—TWV* shows the general slope, from which it is apparent that, in the vicinity of New York City, cold days (+14 F or below) are not generally windy and windy days (22 mph or higher) during the months of December, January and February are seldom extremely cold. In other words the highest wind velocities and the lowest temperatures do not ordinarily occur on the same days. However, the average wind velocity for the cold days was slightly higher than the average for the months of December,

January and February, the former (according to Table 1) being about 15 mph, and the latter (according to Table 2, Chapter 2, A. S. H. V. E. GUIDE, 1933) being 13.3 mph.[3] The average of the mean and minimum temperatures (Table 1) was 12.3 F and 4.4 F, respectively.

The design conditions for New York City would ordinarily be the average wind velocity of 13.3 mph and a temperature 10 or 15 deg above the lowest on record, or, for 15 deg, -6[4] $+ 15 = +9$. These conditions compare with the average wind velocity of 15 mph and the average mean and minimum temperatures of 12.3 F and 4.4 F, respectively, shown in Table 1. Averages of means and extremes, however, are likely to give misleading results and hence an

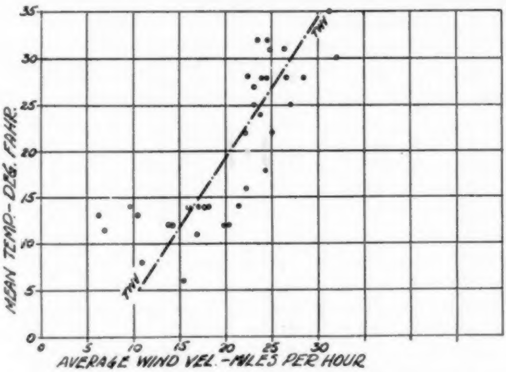


FIG. 4. MEAN TEMPERATURES AND AVERAGE WIND VELOCITIES SHOWN IN TABLES 1 AND 2

analysis was made of the heat losses for the specific conditions shown in Tables 1 and 2.

Constant heat loss curves for Rooms Nos. 1, 2 and 3 are plotted in Figs. 5 and 6. For example, in Fig. 5, all conditions on the 15,000-Btu curve represent a heat loss of 15,000 Btu per hour. Thus, a temperature of about $+2$ F and a wind velocity of 5 mph would result in the same hourly heat loss of 15,000 Btu as a temperature of about 26 F and a wind velocity of 15 mph.

The calculated heat loss of Room No. 1 for the assumed design conditions of $+9$ F and 13.3 mph is 19,800 Btu per hour, which condition is indicated immediately to the left of the 20,000-Btu-per-hour line. The mean temperature and wind velocity conditions given in Tables 1 and 2 and designated by dots on Fig. 5 show that in many cases the heat loss would exceed the design heat loss of 19,800 Btu, for this particular case, and for the assumed equilibrium conditions. It is worthy of note that the curves of Fig. 5 show approximately the same general slope as the temperature-wind velocity curve of Fig. 4; also that in many cases the more severe conditions are represented by higher outside

[3] This average has diminished somewhat in recent years and according to the latest records the average for the three coldest months is 11.3 mph.

[4] -6 F is the minimum temperature for New York City, according to Table 2, Chapter 2, A. S. H. V. E. GUIDE 1933. A lower temperature has been recorded in New York City since this table was compiled.

temperatures at higher wind velocities. In other words, the greatest heat loss would not take place at the lowest mean temperature of + 6 F when the average wind velocity was 15.4 mph, but rather at three other conditions namely 12 F – 20.3 mph, 14 F – 21.4 mph, and 18 F – 24.3 mph.

In the case of the all-brick walls (Room No. 2), the constant heat loss curves for which are shown in Fig. 6, combinations producing the maximum heat losses are quite different. Two temperature-wind velocity conditions indicate heat losses approximately equal to the design heat loss, whereas a third condition, namely + 6 F and 15.4 mph, results in a heat loss somewhat higher than the design heat loss. The average temperature of + 6 F is the lowest given in Table 1. Likewise this temperature produces the highest heat loss for the

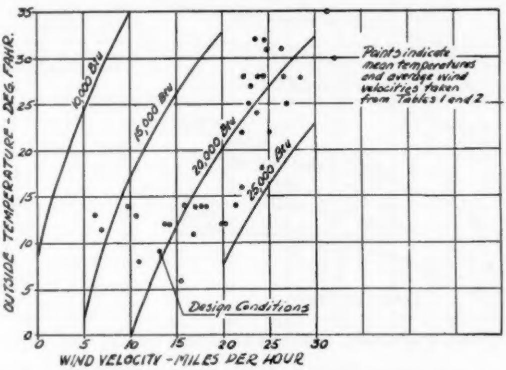


FIG. 5. CONSTANT HEAT LOSS CURVES FOR ROOM NO. 1

glass walls, the constant heat loss curves for which are shown by dotted lines, Fig. 6.

In no case are the design conditions the most severe, and the condition which produces the maximum heat loss in the case of the construction shown in Fig. 1 differs from that which produces the maximum heat loss for the other two types of construction.

The following corollaries may therefore be deduced:

B. For any given enclosed space, some concurring combination of temperature and wind velocity will result in a heat loss equal to or greater than other combinations.

The temperature may not necessarily be a minimum nor the wind velocity may not necessarily be in the highest range; furthermore, the concurring combination of conditions may not necessarily be a temperature of 10 or 15 deg above the lowest on record, and the average wind velocity during the three coldest months of the year.

C. The temperature wind velocity condition to be selected for calculating the maximum heat loss of a building varies with the type of construction, with particular reference (1) to the amount of crack through which infiltration may take place and the relation of the heat loss resulting thereby to the transmission losses and (2) to the ratio of the surface resistances of those parts of the building structure exposed to the wind, to the total resistance of such exposed parts.

In the cases analyzed, it was found that one or more concurring combinations of temperature and wind velocity resulted in a greater heat loss than the assumed design conditions. It seems probable that in other cases, or in other localities, the reverse might be true, because there are indications that occasionally the calculated heat loss is excessive. This may be due to improper selection of temperatures and wind velocities, failure properly to allow for the diffusivity of the structure, or to other causes such as absorption of solar radiation. However, the analysis thus far appears to indicate that improper

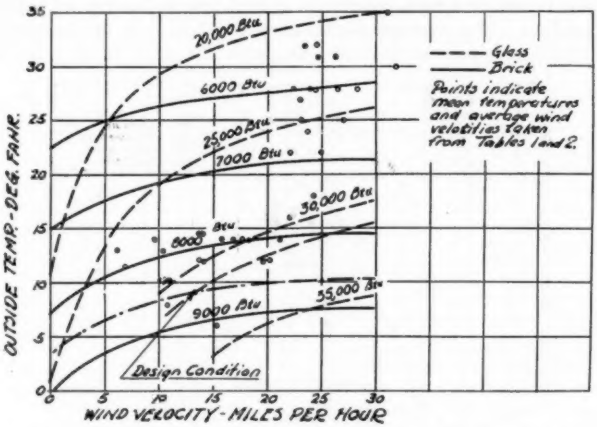


FIG. 6. CONSTANT HEAT LOSS CURVES FOR ROOM NO. 2 (ALL BRICK) AND ROOM NO. 3 (ALL GLASS)

selection of temperatures and wind velocities may result in substantial discrepancies between calculated and actual results.

HOURLY VARIATIONS

Fig. 7 shows hourly heat loss variations for three days for Room No. 1 as well as the means for these days, and the design heat loss for this room based on +9 F and 13.3 mph. The data for the three days considered are given in Tables 3, 4 and 5. January 28, 1925, was selected because it was an extremely cold day for New York City. February 20, 1927, was selected because of the high wind velocity on that day, whereas January 1, 1933, was included because it was one of the most severe days during the past few winters.

If the rate of heat loss reacted instantaneously to the weather changes, the curves would be as indicated. However, on account of the diffusivity of the structure, the actual changes would be less marked and the true curves probably would range somewhere between the theoretical instantaneous curves shown and the means for each day.

Attention is called to the fact that the curves for all three of the days selected have the same general characteristics. The maximum heat loss in each case occurred during the early morning hours, which if generally true, might have

a bearing on the temperature-wind velocity conditions to be selected for making heat loss calculations. In other words, the conditions used might depend on whether the building is to be heated 24 hours per day or just during the waking hours. However, the heating plant probably should be of sufficient capacity to maintain the desired temperature at all times, in case the need should arise.

An inspection of the curves (Fig. 7) would also raise doubt as to the advisability of considering solar radiation in calculating heat losses, since the maximum demand in all cases occurred during the early morning hours when the sun was not shining. Furthermore, the date on which the maximum average heat loss occurred, namely Feb. 20, 1927, was the exceptionally windy day

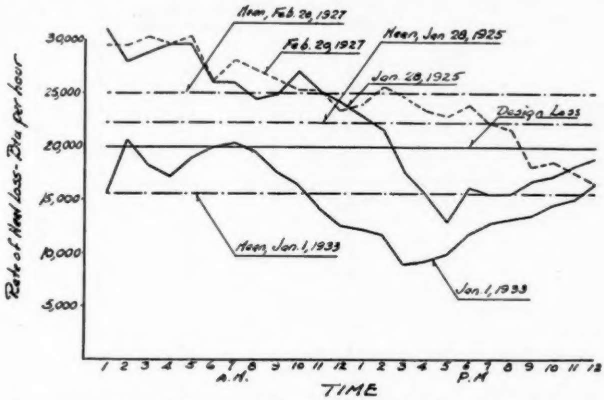


FIG. 7. HOURLY HEAT LOSS CURVES FOR ROOM NO. 1 FOR JAN. 28, 1925; FEB. 20, 1927, AND JAN. 1, 1933

selected, and was cloudy. It will be observed from Fig. 5 that this date was one of extreme heat loss.

Fig. 7 also illustrates why a heating plant, calculated according to present methods, will apparently function satisfactorily for many years with considerable reserve and will then fail on some day of unusual conditions. This in the writer's opinion is due to the fact that present methods of selecting temperatures and wind velocities do not accurately measure the most severe conditions likely to be encountered. While these particular studies show that the design conditions would have been exceeded in several cases, it is not improbable, as previously stated, that in other localities the design temperature and wind velocity based on present methods might result in a heat loss far in excess of that which would ever be encountered.

PERIOD OF CONCURRENCE TO BE CONSIDERED

If a concurring combination of temperature and wind velocity should be selected for calculating heat losses, the question naturally arises as to the period of concurrence to be considered. In other words, should the mean temperature and average wind velocity selected be for a 24-hour period, for a 1-hour period, or for some other interval of time.

Extreme weather conditions are usually of short duration and are seldom repeated on successive days. Furthermore, it is a well-known fact that the diffusivity of the structure tends to bridge-over temperature valleys of short duration. Hence, it would appear that the period of concurrence to be considered in any case would depend upon the duration and severity of the temperature-wind velocity combination and the diffusivity of the structure. The diffusivity, however, has no direct bearing on the infiltration losses, and relates

TABLE 3. HOURLY TEMPERATURES AND AVERAGE WIND VELOCITIES FOR JANUARY 28, 1925, IN NEW YORK CITY[b]

Time	Temperature	Average Hourly Wind Velocity	Direction of Wind
1 a.m.	4	26	N N E
2	4	22	N N E
3[c]	1	21	N N E
4	0	22	N N E
5	−1	21	N N E
6	−2	17	N N E
7 a.m.	−2	17	N N E
8	−1	15	N N E
9	2	17	N N E
10	5	21	N N E
11	6	19	N N E
12	7	18	N E
1 p.m.	10	18	N E
2	11	17	N E
3	13	12	N E
4	13	9	N E
5	11	5	N N E
6	11	9	N N E
7 p.m.	11	8	N N E
8	11	8	N N E
9	11	10	N N E
10	12	11	N E
11	13	13	N E
12	13	14	N E

[b] Data from New York Meteorological Observatory.
[c] Slight precipitation about 3 a. m.
Max. velocity 34 mph N N E.

only to the transmission losses. The study thus far would seem to indicate that a 24-hour rather than an hourly period should be considered.

EXPOSURE FACTORS

The present practice is to apply an exposure factor to the wall and glass transmission losses and to the infiltration losses on the sides of the building exposed to the prevailing winds. Table 1 shows that on cold days the prevailing direction of the wind was usually northwest or north-northwest. However, in the case of the windy days (Table 2) the prevailing direction of the wind was not as well defined, although the northerly directions predominated. The heat losses on the windy days were frequently as high or higher than on the

cold days, and yet from a study of Table 2 there is some doubt as to the direction to which such exposure factors should be applied, unless specific conditions are taken. Apparently, the only direction which would be excluded would be the south.

Exposure factors were originally applied to allow for incomplete data or ignorance concerning the solution of certain phases of the problem. If the temperatures and wind velocities are properly selected and the coefficients and other constants used are accurate, there appears to be no reason for using

TABLE 4. HOURLY TEMPERATURES AND WIND VELOCITIES FOR FEBRUARY 20, 1927, IN NEW YORK CITY[b]

Time	Temperature	Average Hourly Wind Velocity	Prevailing Direction
1 a.m.	24	39	N E
2	24	39	N E
3	23	39	N E
4	23	38	N E
5	23	39	N E
6	23	33	N E
7 a.m.	24	37	N E
8	25	36	N E
9	25	34	N E
10	25	32	N E
11	25	32	N E
12	25	29	E N E
1 p.m.	25	30	E N E
2	25	33	E N E
3	25	31	E N E
4	25	29	E N E
5	25	28	E N E
6	25	30	E N E
7 p.m.	25	27	E N E
8	25	26	E N E
9	25	20	E N E
10	25	21	E N E
11	25	19	E N E
12	25	17	E N E

[b] From New York Meteorological Observatory.
Note: Percent of maximum possible sunshine —0 percent.
Precipitation, 0.4 in.; rained all day.

exposure factors. The allowance for wind direction could be made by using prevailing wind velocities on the sides exposed to the prevailing winds and non-prevailing velocities on the side or sides exposed to non-prevailing winds.

CONCLUSIONS

1. Transmission losses depend upon the temperature head, and to a minor extent (except in the case of highly conductive walls), upon the wind velocity; infiltration losses depend upon both the temperature head and the wind velocity. Since the proportions of transmission and infiltration losses to the total vary with the type of construction and temperature and wind conditions, there is no

general correlation between temperature and wind velocity for calculating heat losses.

2. For any given room or space, some concurring combination of temperature and wind velocity will result in a heat loss equal to or greater than other combinations.

3. The temperature-wind velocity condition to be selected for calculating the

TABLE 5. HOURLY TEMPERATURES AND AVERAGE WIND VELOCITIES FOR JANUARY 1, 1933, IN NEW YORK CITY[b]

Time	Temperature	Average Hourly Wind Velocity	Direction of Wind
1 a.m.	25	15	N N W
2	22	22	N N W
3	20	17	N N W
4	18	14	N N W
5	17	16	N N W
6	16	17	N N W
7 a.m.	15	17	N N W
8	13	15	N N W
9	13	12	N N W
10	15	11	N N W
11	18	9	N N W
12	22	8	N N W
1 p.m.	23	8	N N W
2	25	8	N W
3	27	4	N W
4	28	5	W N W
5	27	6	W S W
6	27	9	W S W
7 p.m.	27	11	W S W
8	27	12	W S W
9	26	12	W S W
10	26	14	W S W
11	26	15	W S W
12	25	17	W S W

[b] Data from New York Meteorological Observatory.

heat losses varies with the type of construction, and for any individual case may be determined from an analysis of local weather conditions.

4. It is probable that temperature-wind velocity combinations for design purposes may be readily ascertained for each locality and for one or more common types of construction, with particular reference to the approximate ratio of the infiltration loss to the total loss.

5. In most cases solar radiation should be neglected since the maximum heat loss usually occurs in the early morning hours when the sun is not shining.

DISCUSSION

A. C. WILLARD: I am very much interested in the paper that has just been presented. The philosophy underlying this paper is not new, nor is it novel in any way,

but it serves one admirable purpose. It brings together for the consideration of this Society in one document practically all those factors which we have discussed during the life history of this Society, usually as separate items in separate papers.

I think that Mr. Bolton 30 or 40 years ago pointed out, just as the author has indicated here in one connection, that a high outside temperature may, when accompanied by a very high wind velocity, prove much more disastrous from the standpoint of heating and impose a much heavier load than low temperatures and little wind velocity.

Mr. Close has gone further in bringing together all these combinations of factors in an analytical study, which brings us back to where we were many years ago. We have done very little to verify Mr. Close's predictions. He was modest in his role of prophet at the conclusion of his paper, but it was merely a prophecy and we have a most unfortunate lack of correlating data or data which we can correlate with these analytical studies.

As was announced by Prof. G. L. Larson, Chairman of the Research Committee, the latest attempt of the Society is to make such a study over at least the central part of the United States. An endeavor will be made to bring together some of the facts that are essential to any one in adjusting the theoretical or scientific data, as determined from various laboratory studies of conductivity, infiltration and such other factors, to bear on the subject of the actual heat loss that takes place.

The last slide shown on the screen by the author demonstrates how necessary those data are. You will notice he had a very definite set of curves which the mathematician and the analyst can develop to show the heat loss that would take place (and I wish to stress this tremendously) if the instantaneous heat loss followed the instantaneous temperature and wind movement. If that is true, Mr. Close's paper is the final answer to this whole question, but, gentlemen, if that isn't true, we are guessing just as much as though we didn't have any of these basic data. We must know something more than we do know at present about the correlation of practice; that is, the results under actual operating conditions, with these very accurately determined fundamental figures, or coefficients, or factors, that are obtained from laboratory studies. Please understand this is no reflection on the laboratory studies, but what we lack are these correlating factors.

In such laboratories as Dr. C.-E. A. Winslow has discussed, and others in the country, such as the Society's Laboratory at Pittsburgh and several university and commercial laboratories, we can secure a great field of laboratory data; there is no question about that, but somebody has to apply those results. This Society has developed an enormous amount of basic material of this theoretical nature based on very clever analysis, but its application still leaves a great deal to the imagination and to the determination of the significance of *if such and such a condition is true*. The author has pointed out very thoroughly all these relationships, but he lacks, and we all lack, these correlating data.

W. L. FLEISHER: I would like to say a few words on the subject of wind directions. I suppose it is well known that the *Heating and Piping Contractors National Association* for years made a study of wind velocities and directions in connection with prevailing temperatures, and an enormous amount of data were collected for 40 or 50 different cities in the United States. It was very interesting to note during this investigation that the wind direction varied so much in prevailing direction in different sections of the country that a prevailing direction could not be assumed.

I think it is pretty well remembered also that the *Heating and Piping Contractors National Association* in working up their method of figuring radiation assumed one mile an hour as equivalent to a one-degree drop in temperature and there was then eliminated from their records about 10 or 12 hours per month of the lowest equivalent conditions which would have corresponded to a lower temperature. From this a base temperature for each locality was calculated and then with the prevailing winds or the direction of the wind, they worked out exposure factors, which have

been used by the heating contractors for about 10 or 12 years. It is from this type of information that Mr. Close has worked up his correlated data for figuring radiation.

Furthering Professor Willard's remarks, the *Heating and Piping Contractors National Association* appropriated a certain amount of money to test the conclusions they had arrived at relative to the loss of heat due to wind velocity and direction, and a series of tests was made over a year or two in one of the buildings of the New York University group in New York City. The results were so peculiarly different from what had been expected that they were not published. In certain cases where it had been assumed that the direction of the wind would create the necessity for additional heat at one end of the building, it was found that it created the necessity for additional heat at the opposite end of the building, and not on the side of the building against which the wind was blowing. The results were so contrary to expectation, that to take a paper such as has been presented and attempt to figure from the theoretical data without further experimental test is, in my opinion, a dangerous procedure.

I agree with Professor Willard that although Mr. Close has done a very nice theoretical job, it isn't safe in any way to assume that that type of calculation can be used in figuring radiation, without a great deal of additional experimental work.

P. D. CLOSE: In regard to the one mile per hour method used by the *Heating and Piping Contractors National Association*, I attempted in my analysis in the paper to show where in certain cases this arbitrary rule might result in a serious error. We know that wind velocity has a decidedly different effect in the case of different types of construction; it has one effect on constructions having windows and doors, etc., and another entirely different effect on constructions without doors and windows. Any increase in wind velocity has comparatively little effect upon the heat losses in the latter case.

To cite an extreme example to show how this rule might be inaccurate: If a wind velocity of one mile per hour were equivalent to one degree temperature drop, a temperature of 70 deg and a wind velocity of 70 mph would theoretically be equivalent to zero degrees and no wind movement. This, of course, is absurd.

MR. FLEISHER: I would like to say one more word, because it has come up this year so forcibly in the work on THE GUIDE. There seems to be a method or an idea of figuring heating on mean conditions. I think that figuring fuel consumption on mean conditions is the correct thing to do, but to figure heating on mean conditions is dangerous. The contractor is installing heating for the severest weather and, if he figures mean conditions, he may have a period in which he is going to have underheating. I have noticed throughout our records that we are always calculating mean conditions, but there are numbers of days in succession in which those conditions are exceeded by a large amount. Nevertheless, strange as it may seem, even when figuring on mean conditions, very few of our installations fall below the required level. A criticism of our methods is that in figuring mean conditions even on the severest days we fulfill our guaranty. This is probably due to the fact that our calculations are not accurate. I think Mr. Close has brought that out. This question of one degree temperature drop for one mile wind was, it is true, a rule of thumb method, but it was the best method available at the time it was produced. This again strengthens Professor Willard's remarks that we have a great deal of experimental information to obtain in order to learn the correct method of figuring radiation.

I have a theory that we should put in a much smaller amount of radiation than we have ever done before and supplement it in some way by a higher air velocity over a section of the radiator, so that on the severest days we can get more out of the apparatus than we put in, rather than putting in large apparatus to take care of extraordinary conditions.

No. 974

RADIATION OF ENERGY THROUGH GLASS

By J. L. Blackshaw * and F. C. Houghten † (*MEMBERS*), Pittsburgh, Pa.

DURING discussion at the annual meeting of the American Society of Heating and Ventilating Engineers held in Pittsburgh in 1931, an interesting point was brought up concerning the transmission of heat through glass by radiation from low temperature surfaces. Surprising to many was the statement that radiation from such comparatively low temperature sources as steam or hot water radiators and the surfaces of room furnishings does not pass through glass. This idea appeared to contradict earlier Laboratory data,[1] which showed that glass absorbs only about 10 per cent of the solar radiation incident to it, and to suggest that the glass windows of a building would act as heat traps because they would allow most of the radiant energy from the sun to pass into the building, but would not permit radiant energy from inside surfaces to pass out.

At the next annual meeting, a paper by Miller and Black [2] was presented supporting the points brought up a year earlier, giving the underlying physical data of radiation and concluding that there is practically no loss of radiant energy from low temperature sources directly through windows.

At the Cleveland Heating and Ventilating Exposition, the Laboratory demonstrated with its pyrheliometer the variation in radiation intensity caused by different temperatures of the source, and the effect of intervening glass in obstructing such radiation. It was suggested that this set-up was adaptable for further check on the absorbing qualities of glass for low temperature radiation, and the Laboratory was asked to make such a study.

The pyrheliometer and auxiliary equipment used by the Laboratory in this study are shown in Fig. 1. The pyrheliometer, which is fully described in a previous Laboratory paper,[3] has as its heat sensitive element a light weight, blackened copper disc, to which the hot junctions of a thermocouple series are attached, their cold junctions being clamped in the flanged edge of a surround-

* Research Engr., American Society of Heating and Ventilating Engineers Research Laboratory.

† Director, American Society of Heating and Ventilating Engineers Research Laboratory.

[1] Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. Transactions, Vol. 36, 1930, p. 137.

[2] Transmission of Radiant Energy Through Glass, by R. A. Miller and L. V. Black, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 63.

[3] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 231.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by J. L. Blackshaw.

ing brass cup, kept at constant temperature by a water bath. A water-cooled shield in which a small aperture was centered was placed between the radiating source and the pyrheliometer. This aperture and a second one close to the sensitive disc limited the amount of energy reaching the disc to a small cone. Glass or opaque slides could be slowly moved across the cone, close to the water-cooled shield, but between it and the sensitive disc, so that insufficient time was allowed for the small portion of exposed glass to warm up and re-radiate to the sensitive element. All surfaces between the water-cooled shield and the sensitive disc were smoked to minimize the reflection of radiated energy. Hence, only that portion of the total radiation from the heated surface which passed through the apertures and was not absorbed by the glass reached the sensitive disc.

In the test set-up of Fig. 1, the source of radiant energy was a 6-in. square of ¼-in. unfinished boiler plate, which could be heated by an electrical element

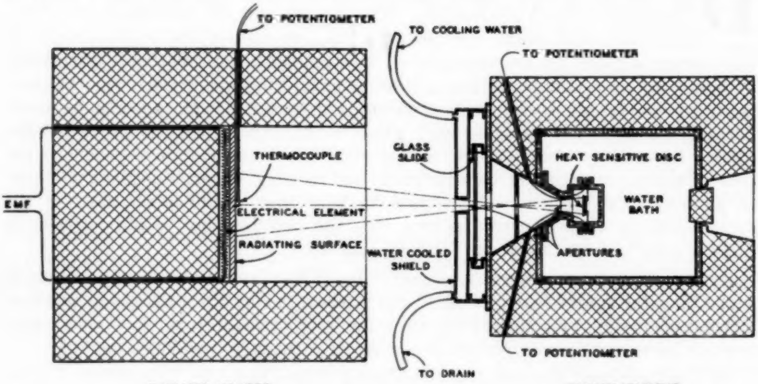


FIG. 1. APPARATUS FOR DETERMINING TRANSMISSION OF RADIATION THROUGH GLASS

to any constant temperature up to a ruddy glow, or 1200 F. Since ordinary wall surfaces and objects found in a room radiate approximately as black bodies, a rough steel surface was used as a source of energy, because work[4, 5] done on such a surface has shown its emissivity to be about 95 per cent of that of a black body. The heat emission in Btu per square foot per hour was calculated from potentiometer readings of the pyrheliometer through the use of a calibration curve derived from the calculated heat emission of the steel plate at different temperatures. This calibration checked to within 19 per cent an extrapolation of a previously established calibration curve for high temperature radiation from the sun determined with a Smithsonian Institution silver disc pyrheliometer. This difference may be accounted for by a change with time in the calibration of the pyrheliometer, or by an error in extrapolating the solar radiation curve to permit its use for low temperature radiation.

For other tests, an electric arc and banks of incandescent tungsten and carbon

[4] Transmission of Heat Through Insulation, by R. H. Heilman. *Mechanical Engineering*, July, 1930, Vol. 52. No. 7, p. 693.
[5] Mark's *Mechanical Engineers Handbook*, 3rd Edition (1930). Table 11, p. 409.

filament bulbs were used as energy sources. When incandescent bulbs were used the apparatus was so arranged that the heat sensitive disc was sighted on a group of the filaments. For these other tests the calibration curve could not be employed. No attempt was made to take into account the wave length distribution of the radiation from different sources.

In order to eliminate stray radiation when making a test, the room in which the equipment was located, the water-cooled radiation shield, and the bath of water in the pyrheliometer were all held constant at 80 F. When the temperature of the radiating source had been adjusted and maintained at the desired point, a routine of tests was made: First, a blackened opaque shield gave a

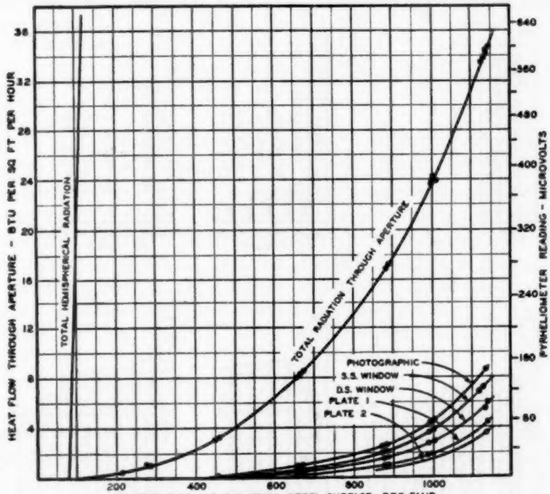


Fig. 2. Relation Between Temperature of Rough Steel Radiating Plate and Radiation Through Unobstructed Aperture and Through Various Kinds of Glass. Receiver and Surroundings at 80 F

check on the zero of the instrument; then, the shield was removed to permit a reading of the total radiation from the source; next, various samples of glass were slid behind the aperture, and potentiometer readings of the pyrheliometer were taken to indicate the radiant energy which passed through the glass. Periodic tests with opaque slides and with unobstructed radiation kept a check on the instrument. Table 1 lists the various types of glass samples used.

Results of tests made with hot rough steel as a source of radiation are given in Fig. 2, where the temperatures of the steel in degrees Fahrenheit are plotted as abscissae and rates of radiant heat flow to the disc, for various types of glass, are plotted as ordinates in Btu per square foot per hour. Potentiometer readings, in microvolts, are shown on an auxiliary ordinate scale. The results given in Fig. 2 are for test conditions where only a part of the total radiation from the surface passed through the aperture in the water-cooled shield, and the

Btu values plotted apply only to the particular arrangement of the equipment and apparatus used in these tests. The curves in Fig. 2 are all based on an absorbing surface temperature of 80 F. It may be seen from these curves that no radiation passed through any of the types of glass tested until the radiating surface had reached 410 F, when a perceptible amount began to register through the photographic specimen. Radiation did not begin to pass directly through plate glass until a temperature of about 550 F was reached. At 1000 F, when the radiating plate was just beginning to glow, it may be seen from the curves that there was no great amount of energy transmitted by any of the glasses.

The almost vertical curve in Fig. 2 shows the total hemispherical radiation

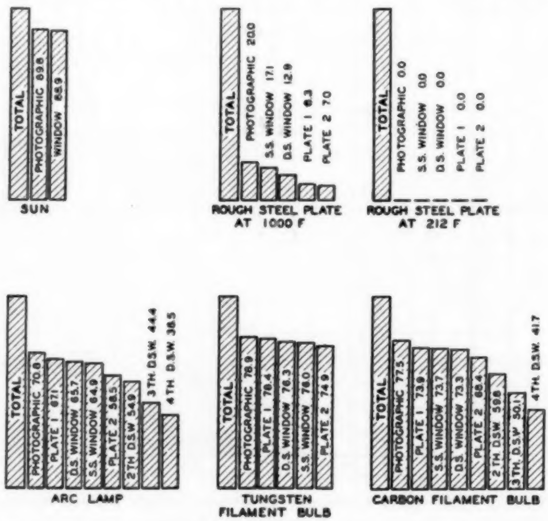


Fig. 3. Comparison of Total Radiation from Different Sources with Percentages Transmitted by Various Kinds of Glass

from the hot plate in Btu per square foot per hour for the different temperatures. This curve was calculated from the Stefan-Boltzmann law with an assumed emissivity constant for the steel surface of 0.95. At a steam radiator temperature, 212 F, the total hemispherical radiation is 190 Btu per square foot per hour, but at this temperature there was no evidence of radiation passing through the glass samples studied.

Data from Fig. 2 may be easily converted to percentages of the total energy radiated through the glass. Such percentages show that when the steel was glowing at a temperature of 1100 F, only 23 per cent of the total radiation passed through the thinnest of the glasses tested. The radiant rays all fell within 4 deg of the normal to the glass surface. Had they fallen at a greater angle, the transmission of energy through the glass would have been even less.

Column diagrams showing the results of tests made with different sources of radiant energy are given in Fig. 3, where the total radiation from each source is compared with the percentage which was transmitted through the respective types of glass. Additional data are shown in Fig. 3 for two, three, and four thicknesses of double-strength window glass when the arc lamp and the carbon filament bulbs were used. Data from earlier Laboratory papers,[1, 3] included in Fig. 3 show the percentage of solar radiation transmitted through glass.

TABLE 1. GLASS SAMPLES

Description	Thickness Inches	Remarks
Photographic	0.065	Clear and colorless
Single strength window	0.080	Greenish tinge
Double strength window	0.125	Greenish tinge
Plate 1	0.2655	Yellowish tinge
Plate 2	0.2390	Greenish tinge

It may be seen from Fig. 3 that the sun, the electric arc, and the electric filaments, which are sources of short wave length, high temperature radiant energy, passed the main part of their energy directly through the glass, while the heated rough steel plate, simulating conditions of low frequency radiant energy in the infra-red range, did not transmit a great amount of energy through glass. Because the steel surface transmitted no registerable energy at temperatures less than 410 F, it is deducted that no appreciable energy radiated from heating surfaces or from the furnishings of a room would pass directly through its glass windows.

Study of Fig. 3 indicates a close check between percentage of radiation for both types of incandescent bulbs. This is probably true because energy is being re-radiated from the heated glass surfaces of the bulbs, and the greater actual temperature of the tungsten filament does not have its relative effect on the total transmission. In addition, it is deducted that most of the energy from the filament that passes through the glass of the bulb itself will necessarily pass through other glass, and that, therefore, the proportional transmission from an incandescent bulb is greater than that from an electric arc.

Comparing the relative amounts of energy transmitted, as shown in Fig. 3, with the varying thicknesses of the respective glasses, listed in Table 1, it may be seen that absorption by glass is not proportional to the thickness of the glass.

SUMMARY

The conclusions of Miller and Black,[2] that glass does not transmit low temperature radiation, are borne out. Ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower. It transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively. This same glass transmits 76.3 per cent of the radiation from an incandescent tungsten lamp, 65.7 per cent of the radiation from an arc lamp, and 88.9 per cent of the radiation from the sun. Thus, glass windows in a room are shown to constitute heat traps, which

allow rather free transmission of radiant energy into the room from the sun to warm the objects in it, but do not allow the transmission of re-radiated heat from these same objects.

DISCUSSION

F. G. HECHLER[6] AND E. R. QUEER (WRITTEN): It has, of course, long been well known that glass possesses highly selective properties so far as the transmission of radiant energy is concerned. The authors' results are interesting because they present in a striking way the quantitative data for a range of temperatures of the radiating body from that of a hot water radiator to the sun. The conclusion that window glass transmits no measurable amount of energy radiated from a source at 500 F or lower is correct, but even at the risk of pointing out what should be obvious to everyone familiar with heat transfer, attention should be called to the fact that this does not mean that window glass is a perfect insulator, nor that it is even moderately effective in preventing heat losses from a room. In fact in this regard it is considerably poorer than a sheet of aluminum foil. At low temperatures the glass does not transmit the energy as radiant energy. The energy that is not reflected at the surface is absorbed by the glass raising its temperature until the heat dissipated from both surfaces by convection and radiation is equal to the heat absorbed. It would perhaps be more accurate to say that glass is a radiant energy trap rather than a heat trap. Windows are quite cold during the heating season, and become excellent receivers of radiant energy from warmer surfaces within a room. An analysis of window heat losses indicates that about 55 per cent of the heat transmitted is supplied by radiant energy. Thus it is considered inadvisable to place radiant heating units so that they *see* exposed glass. The glass does prevent direct transmission of the radiant energy, but does not serve as a heat trap as might be inferred.

If the solar radiation transmitted by the glass falls on a polished metal surface such as aluminum foil a large part will be reflected. If the reflected energy again impinges on the glass it will be transmitted with little loss. In this case the glass would be neither a heat nor a radiant energy trap. This screening or sifting property of glass absorbing only certain wave lengths is alluded to by the authors in discussing the relative transmission from an incandescent bulb and from an electric arc.

The ability of glass to serve as a radiant energy *trap* and of aluminum foil to reflect radiant energy back through glass was strikingly shown by the comparative data obtained in 1932 at The Pennsylvania State College on the absorption of heat from solar radiation.[7] Four heat flow meters had one face in contact with an aluminum plate and the other side exposed to the air in a box containing crushed ice. A 3-in. air space separated the aluminum plate from an outer test panel on which the sun's rays impinged. The material of this panel as well as its color could be changed as desired. The reference section had the upper and lower surfaces of both the outer and inner test panels painted black. When a piece of double strength window glass was used for the outer test panel, the heat flow recorded by the heat flow meter was 240 per cent of that given by the reference section. When the receiving surface of the lower panel was changed from black paint to aluminum foil, the heat flow dropped to 88 per cent of that for the reference section. This shows how effectively the aluminum surface reflects the radiant energy transmitted by the glass. This property might be utilized by putting aluminum foil on the window side of shades at sunny windows to reflect a part of the incident radiation back through the glass. Incidentally, such a shade would also be more effective in winter

[6] Pennsylvania State College.

[7] Surface Absorption of Heat from Solar Radiation, F. G. Hechler and E. R. Queer, *Refrigerating Engineering*. February 1933.

in conserving heat than the usual type; this is due to the low emissivity of aluminum foil for both solar and ordinary temperature radiation.

A few results showing the effect of various kinds of surface treatment are shown in Table "A." Changing the receiving surface of the absorbing panel from black shellac to aluminum foil greatly reduces the heat absorbed; however, a simple treatment of the outer surface of the glass with a coat of white lacquer or white-wash is even more effective. From a practical standpoint white-wash has many advantages for use on skylights and similar exposed glass areas. It is cheap and effective in reducing glare and the transmission of solar radiation to the interior, and is gradually washed away so that by winter, when maximum light transmission is required, it is practically gone.

TABLE A. ABSORPTION OF SOLAR RADIATION, DOUBLE STRENGTH WINDOW GLASS

Receiving Panel		Absorbing Panel		Heat Flow Btu/hour (Sq Ft)		Per Cent Standard[a]	Temperature		
							Surface		Air (Shade)
Top	Bottom	Top	Bottom	Test	Standard[a]		Test	Standard[a]	
Clear	Clear	Black	Black	90.0	37.3	240	101.9	119.9	82.2
Clear	Clear	Aluminum Foil	Black	29.5	33.5	88	87.9	116.7	86.3
White Wash	Clear	Aluminum Foil	Black	21.1	36.7	58	75.3	118.4	78.9
White Wash	Clear	Black	Black	41.4	40.4	102	80.8	129.0	85.2
White Lacquer	Clear	Aluminum Foil	Black	23.3	34.7	67	88.1	114.1	82.8
White Lacquer	Clear	Black	Black	39.5	41.1	96	84.5	123.0	83.8

[a] The standard section had a receiving panel of sheet iron painted black on both sides with a 3 in. air space separating it from an absorbing panel painted black on both sides.

R. A. MILLER: I didn't entirely understand the problem in hand when I supplied the author with glass. The visible light transmission of glass, plate glass, and clear window glass, is approximately 90 per cent in the visible portion of the spectrum. Taking the entire solar spectrum, including the ultra-violet and infra-red, ordinary plate glass will transmit 77 per cent of the total solar energy.

In discussing the transmission of infra-red through glass, there are several factors in the composition of the glass which can affect the total transmission of that glass. If the iron content in the glass is relatively high, the visible light transmission is somewhat reduced, the ultra-violet transmission is almost entirely eliminated and the infra-red transmission is rapidly cut down. So it is possible to obtain glass ⅛-in. thick which will cut out approximately 42 per cent of the incident infra-red, approximately 20 per cent of the incident visible light and virtually 100 per cent of the incident ultar-violet.

Several companies making so-called actinic glasses, non-infra-red transmitting glasses, are producing several different types transmitting various proportions of the infra-red. Any one of those glasses by varying its thickness can be made to transmit more or less of the infra-red in direct proportion to the thickness of the glass used. I mean in inverse proportion to the thickness of the glass used.

If the glass thickness is increased, the infra-red absorption will increase steadily. If the glass thickness is diminished, the infra-red transmission will increase steadily.

The possibility of obtaining glass from any glass producer to cut off a definite amount of infra-red or of ultra-violet depends entirely on the thickness of that particular glass which may be specified.

The conclusions arrived at by the Research Laboratory of the A.S.H.V.E. bear out the conclusions which we had previously reported and they can be further amplified by increasing the thickness of the glass. It was noted in the last curve shown, that where glass was superposed one plate upon another and then where a single glass was used, varying the angle of incidence of the sunlight, the absorption rapidly increased. This was due primarily to the increasing thickness of the glass and the corresponding absorption.

In the case of the single glass used, as the angle of incidence was increased, the thickness of the glass through which the radiant energy had to pass was essentially increased in proportion to the angle. The two curves should lie very nearly together as indicated and should be, if the angle is properly measured, almost identical.

W. W. SHAVER[8]: The results of Mr. Blackshaw's work and our own experiments with ordinary window glass check very well.

In the case of ordinary window glass with approximately 88 per cent transmission of the total sun's energy as shown on the curves, the remaining 12 per cent being reflected or absorbed, of that remainder about 8 per cent represents reflection and 4 per cent absorption. That is to say reflection plays an important factor in the dissipation of this 12 per cent of energy.

The absorption of both heat and light will be increased by increasing the thickness of the glass and by changing the composition. In the case of air conditioning problems, where refrigeration is necessary, the most desirable glazing would be obtained with glass that will transmit as much visible light as possible, and dissipate as much infra-red energy as possible. In addition it is important to have the maximum possible heat resistance since a good heat absorbing glass will be subjected to very appreciable thermal shocks in service, owing to the fact that such a glass gets hot in sunlight.

There are now available heat-resisting heat-absorbing glasses which transmit as high as 75 or 80 per cent of the visible light and only about 42 per cent of the sun's total energy. Since approximately 44 per cent of the sun's total energy on a dry summer day is in the visible region it is evident that a glass with the above-mentioned characteristics must absorb a very high percentage of the sun's infra-red energy, and its efficiency as a window is correspondingly high. In view of these developments I would like to take exception to Mr. Blackshaw's statement that glass is glass.

I would like also to call attention to the remark that glass windows act as heat traps. To a certain extent they do behave as heat traps in that they are opaque to practically all the radiation from the walls and contents of a room. However, the glass windows also radiate energy depending on their temperature and naturally they radiate in all directions, outside as well as inside. Mr. Blackshaw reduced the effect of this radiation from the glass by sliding the plate across the aperture of the furnace and pyrheliometer. In actual service, of course, such means of minimizing the effect of re-radiation from the windows is not possible and hence a window can scarcely be termed a heat trap.

MR. MILLER: If you have two glasses parallel, the outer glass will transmit to the inner glass only the visible energy and the re-radiated energy of the temperature of the glass. Consequently, since the outer glass cannot arrive at the temperature of the incoming energy, the energy re-radiated to the second glass will be of longer wave length and cannot be entirely transmitted through the inner glass.

Essentially, our greenhouses and all of the solariums that we have depend upon the low radiation transmission of glass for the short wave lengths of normal temperature in which we live and move. If the temperature of the glass is kept low, the re-radiation from that glass will be low.

[8] Physicist, Corning Glass Works.

No. 975

STUDIES OF SOLAR RADIATION THROUGH BARE AND SHADED WINDOWS

By F. C. Houghten ** *(MEMBER)*, Carl Gutberlet *** *(NON-MEMBER)* and J. L. Blackshaw ‡ *(MEMBER)*

FORMER studies [1] made at the Research Laboratory of the American Society of Heating and Ventilating Engineers have indicated that solar radiation through windows is a large portion of the cooling load in summer air conditioning. A study,[2] made by Walker, Sanford, and Wells in co-operation with the Society, of the sources of heat comprising the summer cooling load in air conditioning a modern office building has shown solar radiation through windows to be the predominating factor. Consideration of these results by the Committee on Research led to their authorization of the investigation here reported concerning heat gain through windows from solar radiation under different weather conditions, and as affected by several types and applications of window shading appurtenances.

The investigation was made by the A.S.H.V.E. Research Laboratory in Pittsburgh in a two-room test house, Fig. 1 and Fig. 2, built for the purpose on a high, open plot of ground unaffected by shadows. The construction, consisting of board insulation on the outside and inside of 2 by 4-in. studding with a blanket insulation fill, had a wall transmittance of 0.06 Btu per square foot per hour per degree temperature difference from air to air. All joints were caulked to give a low infiltration rate. The east and west walls and the roof were shaded with aluminum painted canvas in order to minimize the effect of solar radiation. The inside walls, ceilings, and floors were painted gray. Dark gray cheesecloth screens were placed within the rooms so as to intercept all direct radiation from the sun through the windows and keep it from being absorbed by the floors, the walls, or the cooling units. Each room

---

** Director, Research Laboratory, American Society of Heating and Ventilating Engineers.
*** Research Assistant, Research Laboratory, American Society of Heating and Ventilating Engineers.
‡ Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers.

[1] Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten, and Carl Gutberlet, A.S.H.V.E. Transactions, Vol. 36, 1930, p. 137.

Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 231.

[2] Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford, and E. P. Wells, A.S.H.V.E. Transactions, Vol. 38, 1932, p. 285.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by F. C. Houghten.

had a window facing due south made of a single sheet of double strength A-quality window glass with a clear unobstructed area of 17.24 sq ft.

Each room contained a commercial ice cooling unit, a small electric heater, a psychrometer with an air circulating fan and with thermocouples to indicate wet- and dry-bulb temperatures, a thermostat, an air circulating fan, two Nicholls heat flow meters, and necessary wiring and piping connections. The temperature in each room was maintained by balancing with the thermostat the cooling unit against the electric heater. The rate of cooling was controlled by automatic intermittent operation of the air circulating fan within the unit and by hand adjustment from outside the rooms of inlet and outlet dampers.

Thermocouples selectively connected to a potentiometer set-up made it possi-

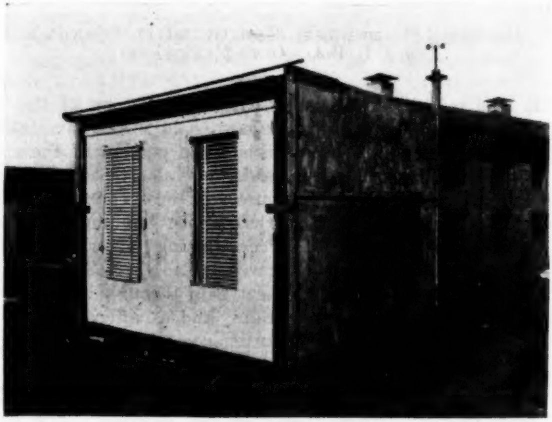


FIG. 1. TEST HOUSE WITH VENETIAN BLIND APPLIED TO INSIDE WINDOW OF EAST ROOM (RIGHT) AND TO OUTSIDE WINDOW OF WEST ROOM

ble to observe (1) air temperatures at various points within the rooms, (2) surface temperatures of inside and outside walls, ceilings, floors, and window glass, (3) outside air temperatures a few inches from the glass, (4) outside air temperatures in the shade, and (5) surface temperatures at desired places on the various window appurtenances used. Nicholls heat flow meters applied to the centers of the inside east wall of the east room and of the west wall of the west room determined the heat flow through the surfaces of these walls. Heat flows through other surfaces of side walls, floors, and ceilings were obtained by a second movable meter in each room.

The rate of cooling supplied to each room was determined by collecting the water produced by the melting ice in calibrated graduates at 15-minute intervals and observing the temperature of this water at the point where it left the rooms. The heat input by the fan motors and electric heaters was obtained from periodic reading of the indicating watthour meters for each room.

Solar intensity was determined by the Laboratory's pyrheliometer, which was directed continuously toward the sun by a clock and trunnion mounting. Wind

velocities were obtained from an electrically recording cup anemometer located above the roof.

Consideration was given to the most desirable temperature to be maintained within the test rooms. At first, tests were made with varying room temperatures which would hold the temperature differential between the inside and outside to that advocated by THE A. S. H. V. E. GUIDE 1933 [3] for summer air

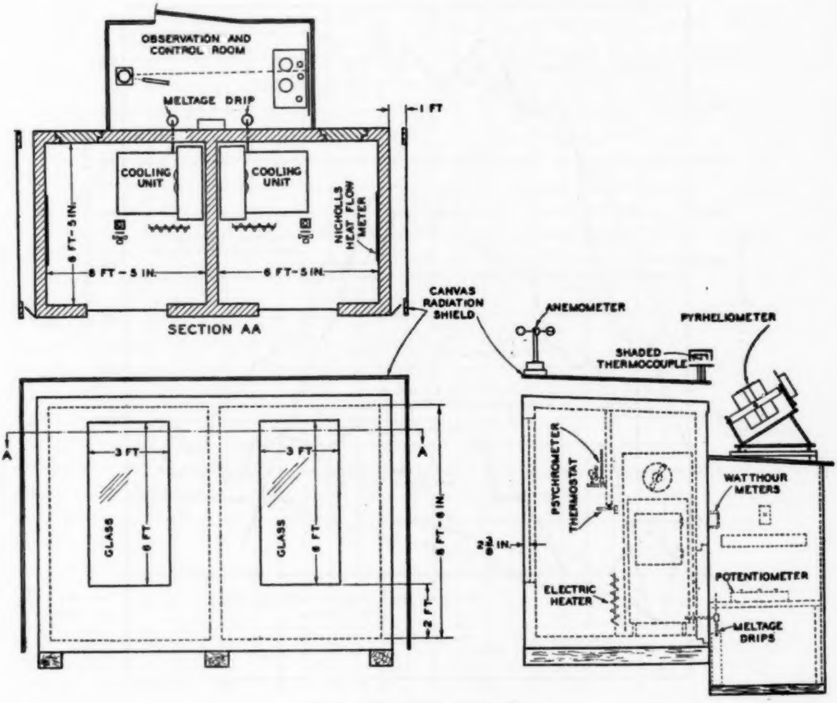


FIG. 2. TEST HOUSE

conditioning. It was soon found, however, that this so complicated the problem that computation would not give satisfactory results, so it was decided to select that temperature which would give the recommended differential at noon and to hold it constant during respective tests. To insure equilibrium conditions of the structure and air during a test, an attempt was made to anticipate this differential to evening before the test and to maintain this room temperature during the night. However, if morning weather conditions indicated that a poor estimate had been made, the air temperature was changed in the direction desired before the sun became effective or the test started, but

[3] Table 2, Chapter 22, A.S.H.V.E. GUIDE, 1933.

this change was never greater than 3 deg in either direction. The ice charge in the cooling units so controlled the relative humidity that the dew-point in the room was always in the neighborhood of 40 F.

In making a test, ice sufficient to last at least 24 hours was put in the coolers the previous evening, the required adjustments to the window appurtenances and to the temperature control equipment inside the room were made, and the

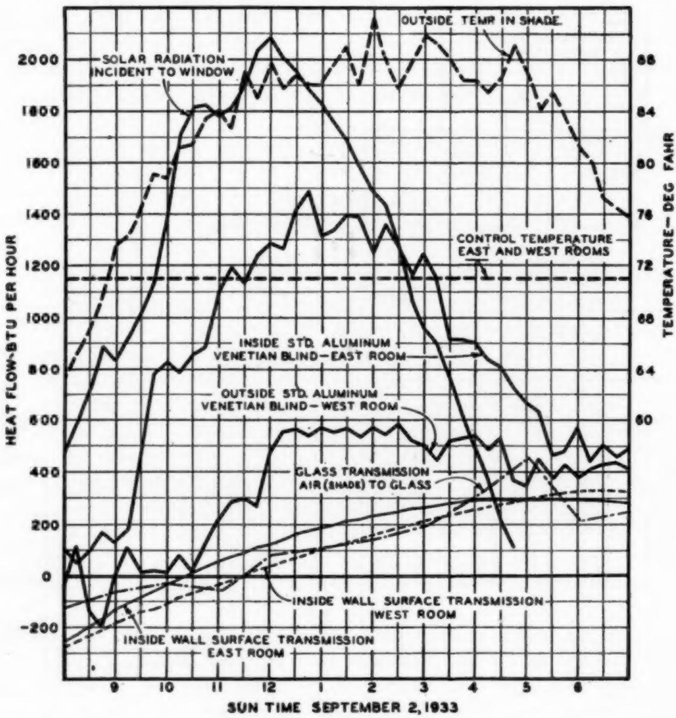


FIG. 3. HEAT TRANSFER AND TEMPERATURE RELATIONSHIPS FOR INSIDE AND OUTSIDE VENETIAN BLINDS

rooms were sealed. Before the sun became effective on the south wall of the building in the morning, the operation of the equipment was checked, the necessary adjustments were made, and the test was started. The test was considered to be ended at any time after the sun ceased to be effective on the south wall or when cooling was no longer required to maintain the desired temperature.

Shades tested were of the ordinary commercial spring roll type made of a filled cotton cloth of medium weight. Their original finish was a buff color, but for some tests they were painted on the side facing the sun with metallic

aluminum in lacquer. In their standard application they overlapped the window frame by approximately 1 in. all around and hung quite snugly to it. For a few tests, wooden cleats were nailed around the shade to hold it so tightly to the frame that there was a minimum transfer of convected air; while for one inside test, the shade was hung with an air gap of approximately 1½ in. between its periphery and the frame of the window to give free circulation of convected air. For another inside test, the shade was half drawn.

Commercial Venetian blinds with two different finishes were tested. Some were stained a dark green; and others were finished by sputtering molten aluminum metal over the wooden slats to affix a solid metallic aluminum coating which both reflected and diffused solar energy falling upon it. The slats of all Venetian blinds tested were 2⅜ in. wide, ⅛ in. thick, spaced 1¾ in. apart. A Venetian blind was considered to have a standard adjustment when it hung close to the frame of the window, fully extended, with its slats set at an angle of 45 deg so they excluded the sun's rays. For one test the slats were closed tightly, and for another they were wide open in a horizontal position. Venetian blinds intended for use on the outside were designed with features which enabled them to be hung away from the window, as an awning without sides, at any adjustable angle up to about 30 deg. Several tests were made with such a blind hung as an awning at the maximum angle, but one test was made with the angle adjusted during the day to admit a maximum amount of light without any direct sunshine getting through the sides.

The awnings used in the study were of a commercial type having sides and an 8-in. decorative fringe, made of medium weight striped canvas supported on a steel framework. In their standard application they projected 44 in. from the building and left the tops of the windows at angles of 40 deg. The tops and sides of the awnings were tightly fastened to the walls of the house. For most tests, the awnings were used just as purchased (standard finish), but for one test an awning was given a heavy coat of metallic aluminum in lacquer on its outside surface.

Complete shading was accomplished by four large canvas shields placed in front of the window to shade the opening from solar radiation. The nearest canvas was held about 7 in. from the glass and 5 in. from the wall, and the other three shields were placed successively farther away and were separated by three 4-in. air spaces. This arrangement allowed free air circulation between the canvas and the window and between the different layers of canvas which eliminated both direct radiation from the sun and re-radiation from the canvas nearest the glass. Hence, the only heat transfer through the window should have been that due to conduction from the stratum of shaded air nearest the window.

Test Results and Analyses

Twenty-five successful tests were made on different window arrangements, with results as listed in Table 1. Five tests, listed also in Table 1, were satisfactory in one room only because of unforeseen difficulties which affected the other. Many additional tests were started but discontinued because of unsatisfactory weather conditions.

The results of Test 32 are plotted in Fig. 3. This test was made on September 2, with an aluminum Venetian blind hung inside the window in the

TABLE 1. ANALYSIS OF COOLING LOAD WITH BARE AND SHADED WINDOWS

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
				Solar Radiation Incident to Glass Btu per Hour						East Room								
										Appurtenance				Rate Btu per … Cooling				
Test Number	Date ~ 1933	Max. Outside Temp.	Max Solar Radiation ~ Normal to Sun's Rays ~ Btu per Sq Ft per Hour	Max. ~ Entire Window†	Max. per Sq Ft of Glass	Av.~Entire Window *	Av per Sq Ft of Glass *	Inside Control Temp.	Av. Air Temp. Difference * Inside to Outside (in shade)	Type	Finish	Location	Adjustment	Max ~ Entire Room	Max. ~ Window Only	Max. ~ per Sq Ft of Glass	Av. ~ Entire Room (9 AM to 5 PM)	Av. ~ Entire Room*
4	July 13	86.7	287	1441	83.6	1072	62.2	75.8	6.3	Complete Shading				420	370	15.7	234	234
6	17	82.4	302	1572	91.2	1029	59.7	76.0	1.3	Shade	Buff	Inside	Cleated	657	489	28.4	419	448
7	18	86.3	224	1104	64.0	677	39.3	76.4	4.4	"	"	"	With Gap	751	693	40.2	514	504
8	19	92.3	259	1374	79.7	946	54.9	76.4	10.4	"	"	"	St'd	943	521	30.2	719	732
9	20	95.6	267	1443	83.7	1024	59.4	76.4	10.8	"	Al.	"	"	1177	733	42.5	825	850
10	21	99.7	254	1385	80.3	970	56.3	76.4	17.4	"	"	"	Cleated	972	652	37.8	719	734
12	23	102.6	252	1386	80.4	977	56.7	76.4	20.9	"	"	Outside	"	584	170	9.9	546	528
13	24	95.4	238	1306	75.7	837	48.6	76.4	13.0	"	Buff	"	St'd.	789	554	32.1	529	571
15	31	97.8	247	1484	86.1	1026	59.5	79.4	14.3	"	"	Inside	Cleated	858	595	34.5	729	742
16	Aug. 1	97.2	243	1483	86.0	991	57.5	81.3	10.8	"	"	"	St'd.	947	610	35.4	784	829
17	4	80.5	311	1940	112.5	1360	78.9	81.3	-7.5									
18	7	89.0	309	2020	117.2	1342	77.8	80.4	3.4	Awning	St'd	Outside	St'd.	534	384	22.3	398	403
20	11	87.9	297	1887	109.5	1283	74.4	76.4	5.5	"	Al.	"	"	515	392	22.7	375	370
21	12	87.9	252	1713	99.4	1243	72.1	77.3	7.0	Ven. Bl.	Green	"	As Awn.	559	434	25.2	481	488
22	15	86.1	281	1994	115.7	1305	75.7	75.3	5.5	"	"	"	"	607	481	27.9	404	410
23	16	89.4	277	1963	113.9	1249	72.4	74.6	7.0	Bare Window				1591	1506	87.4	1066	1102
25	18	85.6	309	2123	123.1	1129	65.5	None	—		"	"						No
26	21	84.9	262	1979	114.8	1389	80.6	76.4	4.4	Ven. Bl.	Al.	Outside	Adjusted	465	383	22.2	356	337
27	22	80.4	301	2149	124.6	1281	74.3	74.6	1.0	"	"	Inside	St'd.	1009	921	53.4	771	771
28	25	90.7	283	2180	126.5	1484	86.1	79.0	6.3	"	"	"	"	1115	988	57.3	819	832
31	30	82.8	286	2349	136.3	1432	83.1	73.0	4.6	Shade	Buff	Outside	"	641	577	33.5	429	436
32	Sept. 2	91.2	245	2093	121.4	1359	78.8	71.0	14.0	Ven. Bl.	Al.	Inside	"	1388	1212	70.3	1019	1030
33	5	87.5	206	1806	104.8	1064	61.7	71.0	10.5									
34	6	90.9	252	2273	131.8	1610	93.4	74.0	11.4	Ven. Bl.	Al.	Outside	St'd.	535	364	21.1	369	369
35	7	92.1	235	2111	122.4	1419	82.0	80.0	6.1	"	"	"	"	374	270	15.7	202	202
36	8	94.5	227	2052	119.0	1394	80.9	83.0	6.7	"	"	"	"	454	319	18.5	268	268
37	9	90.3	212	1969	114.2	1115	64.7	83.0	3.3	"	"	"	"	312	239	13.9	237	237
38	10	81.9	272	2559	148.4	1501	87.1	79.0	-0.5	"	"	"	"	360	266	15.4	276	276
39	17	83.2	265	2670	154.9	1916	111.1	79.0	2.5	Bare Window				2467	2502	145.1	1619	1619
41	Oct 11	71.8	280	3582	207.8	2408	139.7	57.9	9.8	Shade	Buff	Inside	St'd.	1617	1490	86.4	1163	1163
Av All Tests		88.8	265	1913	111.0	1261	73.1	76.5	7.1									

* For period—9 a. m. until the sun ceased to be effective.
† Window glass area—17.24 sq ft.

Abbreviations

With Gap—1½ in. between periphery of shade and frame.
St'd—Standard.
Plain canvas awning.
Shade typically applied.
Venetian blind—hanging down—slats at 45 deg.

TABLE 1 (*Continued*)

20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
					West Room													
5 Hour					Appurtenance				Rates Btu per Hour									
Transmission		Heat Removal							Cooling					Transmission		Heat Removal		
Av ~ Entire Wall *	Av. - Outside Air (shade) To Glass Bare Window Assumed *	Av. Charged to Window (19)-(20) *	Av. Charged to a Sq Ft of Glass *	Per Cent Transmission by Appurtenance $\frac{(19)-(20)}{(7)\div(21)}\times 100$	Type	Finish	Location	Adjustment	Max - Entire Room	Max. - Window Only	Max. - per Sq Ft of Glass	Av. - Entire Room (9 AM to 5 PM)	Av. - Entire Room *	Av. - Entire Wall *	Av. Outside Air (shade) To Glass Bare Window Assumed *	Av Charged to Window (33)-(34) *	Av. Charged to a Sq Ft of Glass *	Per Cent Transmission by Appurtenance $\frac{(33)-(34)}{(7)\div(35)}\times 100$
178	42	36	3.2	5	Bare Window				1361	1366	79.2	1027	1027	-66	42	1094	63.4	98
149	-123	299	17.3	33	"		"		1076	1182	68.6	696	744	-153	-123	897	52.0	99
91	109	413	24.0	53														
187	127	545	31.6	51	Bare Window				1531	1408	81.7	1112	1173	101	127	1072	62.2	100
390	109	460	.26.7	41	"		"		2045	1784	103.5	1229	1310	202	109	1108	64.3	98
276	285	458	·26.6	36	"		"		1868	1643	95.3	1341	1409	207	285	1202	69.7	96
306	381	222	12.9	16	"		"		1910	1658	96.2	1459	1512	215	381	1297	75.2	96
216	273	355	20.6	32	"		"		1780	1560	90.5	1138	1275	151	273	1124	65.2	101
253	262	489	28.4	38	Shade	Buff	Inside	½ Drawn	1219	1022	59.3	1022	1039	170	262	869	50.4	68
312	165	517	30.0	45														
					Awning	St'd.	Outside	St'd.	297	447	25.9	128	121	-169	-223	290	16.8	26
95	-40	308	17.9	24	"	"	"	"	525	521	30.2	334	330	-21	-40	351	20.4	27
79	20	291	16.9	22	"	"	"	"	629	587	34.0	494	456	22	20	434	25.2	33
93	92	395	22.9	30	Bare Window				1775	1696	98.4	1147	1192	-28	92	1220	70.8	91
44	20	366	21.2	28	Complete Shading				196	186	10.8	78	49	-39	20	88	5.1	7
42	-139	1060	61.5	95														
Cooling					Bare Window				No Cooling									
51	-10	286	16.6	21	Complete Shading				349	322	18.7	235	236	166	-10	60	3.5	4
66	-106	705	40.9	60	"		"		120	130	7.5	87	87	27	-106	60	3.5	5
89	-105	743	43.1	54	Bare Window				1784	1810	105.0	1222	1255	-21	-105	1276	74.0	93
48	-5	388	22.5	27	Shade	Al.	Outside	Cleated	332	294	17.1	235	236	6	-5	230	13.3	16
141	127	889	51.6	60	Ven. Bl.	"	"	St'd.	559	465	27.0	374	381	77	127	304	17.6	20
					"	"	"	"	349	299	17.3	263	263	46	158	217	12.6	18
113	182	256	14.8	14	"	"	"	As Awn.	623	508	29.5	541	541	32	182	509	29.5	28
24	40	178	10.3	12	"	"	"	Wide Open	548	462	26.8	375	375	0	40	375	21.8	26
74	52	194	11.3	13	"	"	"	Closed	198	168	9.7	122	122	11	52	111	6.4	8
55	-40	182	10.6	17	"	Green	"	St'd.	283	276	16.0	183	183	36	-40	147	8.5	14
80	-148	196	11.4	15	"	"	"	"	306	328	19.0	189	189	-13	-148	202	11.7	15
29	-221	1648	95.6	97	Complete Shading				138	75	4.4	-87	-87	-159	-221	72	4.2	4
75	138	1088	63.1	43	Shade	Al.	Inside	St'd.	1411	1401	81.3	1081	1081	-13	138	1094	63.5	43
130	54													30	54			

Al.—Aluminum.
Ven. Bl.—Venetian blind.
As Awn.—Venetian blind adjusted away from window, as awning without sides.
Adjusted—Venetian blind used as awning but moved periodically to keep sun off window.
Wide open—Venetian blind—slats horizontal.
Closed—Venetian blind—slats vertical.

east room and a similar blind hung outside the window in the west room. A control temperature of 71 F had been chosen for each room because the outside temperature in the morning was low, but its rapid rise to a high of 91 F at 2:00 p. m. proved this to have been a poor choice, for 78 F would have been in better accord with THE GUIDE recommendation for inside temperature for such conditions. The cooling rate, as shown for each room, reached a maximum of 1480 Btu per hour for the east room, which had the inside blind, and 580 Btu per hour for the west room, which had the outside blind. The negative values occurred when the electrical input required to maintain the room temperature was greater than the heat extracted by the cooler.

Heat flow through the walls, ceilings, and floors of the two rooms calculated from the heat meter readings, and heat transfer from the outside air to the outside glass surface for the prevailing temperature difference and wind velocity for the east window equipped with the inside blind are also plotted in Fig. 3. The great lag in flow through the insulated walls is indicated by the continuance of negative values after the outside air temperature became higher than that inside. Wall transfer to the east room was higher than that to the west room during the early part of the day because the air between the shading canvas and the wall was at a higher temperature on this side of the building during the forenoon.

The maximum rate of solar radiation impingement against the 17.24 sq ft of window area, as determined from the pyrheliometer reading, is 2,093 Btu per hour at noon and falls off rapidly before and after this time because the angle of impingement lessens as the altitude and azimuth angles of the sun change and because part of the glass area is shaded by the frame. The rate of energy impingment on the total area of glass at any instant is given by the formula:

$$E = P\left(\sin\left[\cos^{-1}\sqrt{\{\cos \alpha \cos (90^\circ - \Upsilon + \theta)\}^2 + \sin^2 \alpha}\right]\right)$$
$$\left(wh - hL \tan (\Upsilon - \theta) - wL \tan \alpha + L^2 \tan (\Upsilon - \theta) \tan \alpha\right)$$

where:

E = solar intensity, in Btu per hour, on the window, h feet high, w feet wide, set back a distance L feet from the outside wall surface.
P = solar intensity, in Btu per square foot per hour, normal to the direction of radiation.
α = altitude angle of the sun at the instant considered.
Υ = angle between north and the direction the window faces.
θ = azimuth angle of sun (angle between sun and the north).

The average rate at which energy impinges against the entire glass, given in Table 1 for the period of the day considered, was obtained by integrating the instantaneous rates. This average rate divided by the total glass area gave the average rate per hour per square foot of glass.

The sun became effective on the south window at 7:05 a. m. on September 2 and disappeared at 4:45 p. m. This time of exposure changed from day to day during the summer, being a period of from 8:20 a. m. to 3:40 p. m. on July 15, and one from 6:40 a. m. to 5:20 p. m. on September 15. For the purpose of making a heat balance, the periods from 9:00 a. m. until the sun ceased to be effective were considered, but some values are listed for a longer period, running from 9:00 a. m. to 5:00 p. m., which includes the late after-

noon hours when cooling was required because the high air to air temperature difference continued and because there was a lag in heat flow through the walls.

The average rate of radiation impingement against the south glass, 17.24 sq ft in area, for the period from 9:00 a. m. to the time the sun ceased to be effective on September 2, as given in Table 1, was 1,359 Btu per hour. The effect of altitude and azimuth angles of the sun and of shading by the frame in reducing the solar radiation on the glass is illustrated by comparing this value with 121.4 Btu per square foot (2,093 Btu for 17.24 sq ft) maximum solar intensity, and with 245 Btu per square foot (4,263 Btu for 17.24 sq ft) average solar intensity normal to the direction of radiation.

In analyzing the results of the tests to show the relation of the cooling load to different window conditions and to show the effectiveness of window appurtenances, it was necessary to consider all sources of heat entering and leaving the test rooms. These sources of heat are illustrated in Fig. 4. Solar radiation *A* of a given intensity impinges against the glass surface at an angle

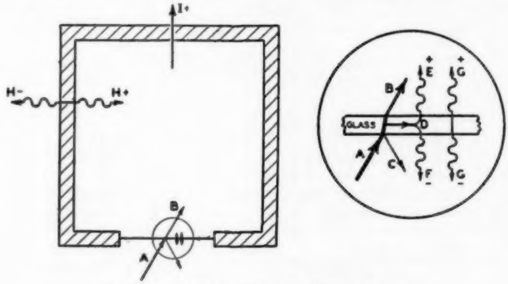


FIG. 4. ANALYSIS OF HEAT GAIN

which changes throughout the day. A part *B* of the radiant energy passes directly through the glass into the room. Another portion of it *C* is reflected from the glass surface, while the remaining portion *D* is absorbed while passing through the glass. The energy *D* raises the glass temperature so that heat *E* is transferred from the inside glass surface to the air inside the room, or heat *F* is transferred from the outside glass surface to the air outside the room, or both. The division of *D* into *E* and *F* is dependent upon the relative temperatures of the glass and of the air on either side. Simultaneously, heat transfer *G* is taking place through the glass between the outside and inside air, its direction and magnitude depending upon the temperature difference. Table 2 shows the temperature gradients obtained in several tests of bare windows, and cites conditions when absorption of radiation in the glass was sufficient to raise its temperature above both the inside and outside air temperature in the shade, or when heat flowed from both surfaces of the window at the same time.

The solar radiation *A*, Fig. 4, incident to the glass plus the conduction *F* between the outside air and the glass surface gives the total heat gain through the window, excepting for the effect of direct reflection of radiation *C* from the outside glass surface. A brief study made of the absorption and reflection of solar radiation by glass, which will be presented later in this report, indicates

that the reflection C is negligibly small. Therefore, in the analysis of the data presented in this paper reflection C is disregarded.

The magnitude and direction of heat flow H through the walls of the room are dependent upon the temperature difference between the air on the inside and that on the outside, but since there is considerable lag in the transfer through a wall, owing to its heat capacity and other factors, the rate of transfer between the inside air and inside surface of the wall and between the outside air and outside surface of the wall may differ widely in magnitude or even in direction at any instant, and may bear little or no relation to the air to air temperature difference. I represents the heat removal from the room occasioned by melting ice.

In order to maintain a constant air temperature within the room, the rate

TABLE 2. TEMPERATURE GRADIENTS AT NOON FROM OUTSIDE AIR IN SHADE THROUGH GLASS TO INSIDE AIR WITH AND WITHOUT SHADING

Test	Date	Temp. in Shade F	Room 1			Room 2		
			Appurtenance	Glass Temp. F	Control Temp. F	Appurtenance	Glass Temp. F	Control Temp. F
4	July 13	81.8	Complete Shading	78.2	75.9	Bare Window	84.2	75.9
27	Aug. 22	82.7	Inside Ven. Blind	84.6	74.7	Complete Shading	76.7	74.7
39	Sept. 17	80.5	Bare Window	85.4	79.0	Complete Shading	80.0	79.0

of heat removal I should equal the summation of all the sources of heat gain. Analysis of the results supported this but showed a small lag; that is, the rate of heat removal from the room always lagged somewhat behind the heat gain. This is demonstrated by the heat removal curves, Fig. 3, which continue after the sun ceased to be effective on the window at a rate greater than the combined rates of heat transfer through the inside wall surface and the glass.

It was desired in analyzing the results of the different tests listed in Table 1 to equate the heat entering through the windows and the heat removal chargeable to the windows in order to establish a balance and account for all transfers.

The calculated heat gain through the window, which is the sum of the radiation impingement on the glass, Column 7, Table 1, and the transfer from the outside air to the glass surface, Columns 21 or 35 (in the absence of a shading appurtenance), is compared with the heat removal charged to the window as determined by the total meltage corrected for electric input, Columns 19 or 33, less the calculated transfer through the walls, Columns 20 or 34. The ratio of the actual heat removal for the respective rooms to the heat which would be gained through an unshaded window is expressed in per cent in Columns 24 and 38.

Averages for all tests made with the same window conditions are given in

Table 3, where five tests with the windows completely shaded have an average percentage ratio of bare window transmission of 5 per cent. This seems a reasonable figure.

The percentages given in Tables 1 and 3 for bare windows represent for eleven tests the heat removed by cooling which was chargeable to the window in relation to that calculated to enter the window. These tests show an average percentage of 97, with a maximum variation of 6 per cent for any test. The discrepancy between these percentages and 100 per cent are chargeable to other sources of heat gain, or to errors in temperature or wind observations on which conductances were based. Probably the largest sources

TABLE 3.—AVERAGE VALUES FOR WINDOW CONDITIONS

Appurtenance†				Number of Tests Averaged	Average Rates of Cooling All Tests Btu per Hour							Per Cent Transmission by Appurtenance	Greatest Variation of Any Test from the Average Per Cent Transmission
Type	Finish	Location	Adjustment		Max. – Entire Room	Max. – Window Only	Max. – per Sq Ft of Glass	Av. – Entire Room (9 AM to 5 PM)	Av. – Entire Room*	Av. Charged to Window*	Av. Charged to a* Sq Ft of Glass		
Complete Shading				5	246	217	11.4	109	104	63	3.9	5	2
Ven. Bl.	Al.	Outside	Closed	1	198	168	9.7	122	122	111	6.4	8	
"	Al.+Green	"	St'd.	9	359	314	18.2	262	263	208	12.1	15	5
Shade	Al.	"	Cleated	2	458	232	13.5	390	382	226	13.1	16	0
Ven. Bl.	"	"	Adjusted	1	465	383	22.2	356	337	286	16.6	21	
Awning	"	"	St'd.	1	515	392	22.7	375	370	291	16.9	22	
Ven. Bl.	"	"	Wide Open	1	548	462	26.8	375	375	375	21.8	26	
Awning	St'd.	"	St'd.	4	496	485	28.1	339	328	346	20.1	28	5
Ven. Bl.	Al.+Green	"	As Awn.	3	596	474	27.5	475	480	423	24.5	29	2
Shade	Buff	"	St'd.	2	715	565	32.8	479	503	371	21.5	30	3
"	Al.+Buff	Inside	Cleated	3	829	579	33.6	622	641	415	24.1	36	3
"	"	"	St'd.	5	1219	951	55.2	914	931	741	43.0	45	6
"	Buff	"	With Gap	1	751	693	40.2	514	504	413	24.0	53	
Ven. Bl.	Al.	"	St'd.	3	1171	1040	60.3	870	878	779	45.2	58	4
Shade	Buff	"	½ Drawn	1	1219	1022	59.3	1022	1039	869	50.4	68	
Bare Window				11	1744	1647	95.5	1187	1238	1182	68.5	97	6

of error involved are the inability to include in the analysis reflection of radiation at the glass surface, and the acceptance of film transfer coefficients for the prevailing wind and temperature conditions used in calculating the heat gain through the windows by conduction.

Similar percentage ratios of bare window transmission are given in Tables 1 and 3 for the windows when affected by various types, finishes, locations, and adjustments of window appurtenances. These percentages seem rather conclusive, and their averages should be acceptable to $\pm$ 5 per cent where several tests were made with the same window condition. This tolerance does not permit

drawing close comparisons between appurtenances of their conditions of application when they show little variation. However, when such tests were made on the same day a direct comparison between the two rooms should be more representative of the actual relationship of the appurtenaces.

TEMPERATURE RISE IN UNCONDITIONED ROOMS

A test was made on August 18 to determine what temperatures would be obtained in the two test rooms on a bright day when no cooling was supplied to either room. The window in the east room was left bare, and that in the west room was completely shaded. Fig. 5 shows that at the start of the day

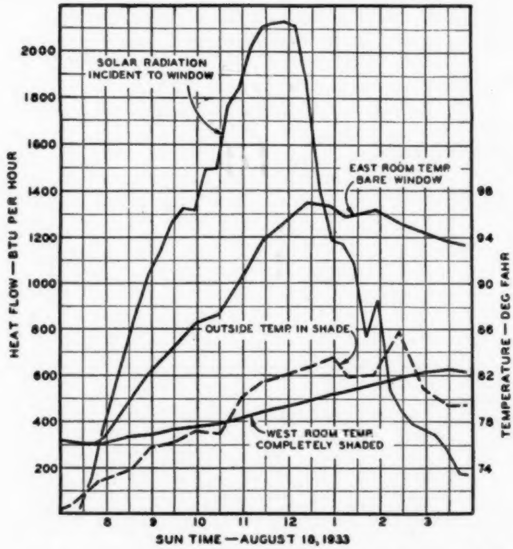


FIG. 5. RELATION OF INSIDE ROOM TEMPERATURES TO BARE AND COMPLETELY SHADED WINDOWS—ROOMS NOT CONDITIONED

each room had a temperature of 76 F, and that while the outside temperature rose to 86 F, that of the unshaded room reached a maximum of 97 F while that in the shaded room was only 82.5 F. It can be seen from the curves that the temperature peak in the shaded room lagged behind the peak in the unshaded room by several hours. Although no quantitative results were obtained from this test, the comparative temperatures found illustrate the desirability of applying shades to the outsides of windows.

INTERCEPTION OF SOLAR RADIATION BY GLASS

During the summer, a related study was made of the comparative percentages of solar radiation intercepted, either by absorption or reflection, by different kinds and thicknesses of glass held at various angles of incidence with the

sun's rays. Table 4 gives the percentage of energy intercepted for the different conditions studied. Curve *A* in Fig. 6 shows the relation between the percentage of radiation intercepted and the thickness of the glass, when one or more pieces of double strength window glass of the same quality as that in the test house were held normal to the sun. Curve *B* in Fig. 6 shows this percentage relation when a single piece of such glass was held at varying angles of incidence with the sun's rays.

No account was taken in this analysis of the refraction of the rays as they passed through the glass. It will be noted that the relation between the radiation intercepted and the thickness of the glass traversed is approximately the same whether the thickness results from placing several pieces of glass normal to the rays, or from changing the angle of incidence of a single piece. This indicates that the increased interception occasioned by changing the

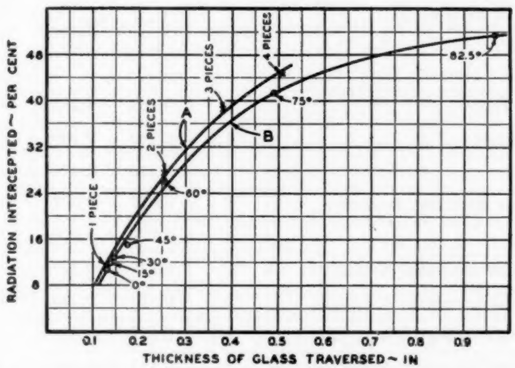


FIG. 6. SOLAR RADIATION INTERCEPTED BY VARIOUS THICKNESS OF DOUBLE STRENGTH A-QUALITY WINDOW GLASS

Curve *A*—Various Pieces Normal to Direction of Radiation.
Curve *B*—One Piece at Various Angles of Incidence with Radiation.

angle of incidence results from increased absorption within the glass because of its greater thickness rather than from the reflection, and that reflection, shown as *C* in Fig. 4, is not an important factor in heat transfer through a window by solar radiation. This assumption was used in the analysis of the data. It should be pointed out, however, that the decreased transmission occasioned by the piling up of several pieces of glass normal to the sun's rays may have resulted from partial reflections at the several surfaces, rather than from increased thickness. It would be of interest to make a more thorough study of the effect had on solar radiation transmission through glass by the thickness of the glass and the angle of incidence, but neither facilities nor time were available for such an extended investigation of this phase of the subject.

AIR MOTION IN THE TEST ROOMS

In order to maintain uniform air velocities in the rooms at all times independent of the rate of cooling required and in order to keep such velocities

within practical limits, an auxiliary fan and housing was located above each of the commercial ice cooling units. Air supplied through the cooling unit by the fan contained therein was discharged into this housing intermittently as the thermostat turned the cooler fan on and off, and at different rates as the inlet and outlet dampers to the units were adjusted. Since the rate of cooling required was always low in comparison with the capacity of the units used, the rate of air flow through the unit was always small. A by-pass around the cooling unit took air from the floor level and admitted it to the housing where

Table 4. Solar Radiation Intercepted by Glass of Various Types, Thicknesses, and at Various Angles

Type of Glass	Thickness			Angle of Incidence Deg.	Radiation Intercepted Per Cent
	Pieces	Each Piece In.	Traversed by Radiation In.		
Plate: Yellow tinge	1	0.255	0.255	0	14.8
Plate: Green tinge	1	0.240	0.240	0	16.6
Plate: Clear	1	0.270	0.270	0	12.8
Window: Single strength; A-quality	1	0.071	0.071	0	10.7
Photographic	1	0.070	0.070	0	9.0
Window: Double strength; A-quality	1	0.127	0.127	0	11.2
Window: Double strength; A-quality	2	0.127	0.254	0	27.8
Window: Double strength; A-quality	3	0.127	0.381	0	38.4
Window: Double strength; A-quality	4	0.127	0.508	0	45.7
Window: Double strength; A-quality	1	0.127	0.127	0	10.8
Window: Double strength; A-quality	1	0.127	0.131	15	11.9
Window: Double strength; A-quality	1	0.127	0.147	30	12.7
Window: Double strength; A-quality	1	0.127	0.180	45	14.9
Window: Double strength; A-quality	1	0.127	0.254	60	26.0
Window: Double strength; A-quality	1	0.127	0.490	75	41.6
Window: Double strength; A-quality	1	0.127	0.969	82.5	53.7

it and the cooler air combined before entering the auxiliary fan which mixed and blew the air horizontally into the room at the 6-ft level. The free delivery capacity of the auxiliary fan was much greater than its capacity to draw air through the by-pass in parallel with the cooling unit. Hence, the air was discharged to the room over a considerable area at a relatively low velocity.

The maximum velocity 30 in. in front of the fan and 3 ft from the wall toward which the air was directed for maximum rate of discharge through the unit was 209 fpm. The maximum velocity as determined by the Kata thermometer 10 in. from the wall toward which the air was directed was 83 fpm. Complete

surveys of the rooms indicated few velocities elsewhere as high as 60 fpm, while at most points near the walls the velocity was found to be less than 40 fpm. Since the Society's ventilation standards approve velocities up to 50 fpm for a well ventilated room, those velocities pertaining in the test rooms were not excessive and should not adversely affect the rate of transfer through any of the walls or windows.

INFILTRATION

Special attention was given to the design and construction of the test rooms to eliminate, as far as possible, the heat gain by infiltration. Every joint in the building, both in the framework and in the insulating board covering all walls, was well caulked. Each thickness of blanket insulation filling the studding spaces was self-contained in tarred paper. It was flanged tightly to the top, bottom, and sides of the studding, with all joints sealed with roofing cement. The entire inside and outside surfaces of the house were painted with three coats of good quality lead paint, care being taken to fill all cracks. The glass was sealed into place with ½-in. moulding strips embedded in putty in the window frames, which were an integral part of the wall construction. The small openings for entrance into the rooms were made of specially flanged construction and were always sealed by rubber gaskets and caulking compound during tests.

Carbon dioxide tests, made in accordance with methods used by Armspach [4] in earlier Laboratory studies, gave infiltration rates of not to exceed 18 cu ft per hour for each of the rooms. For the maximum inside and outside temperature differences and the dew-points experienced in the tests, this rate of infiltration would cause a combined sensible and latent heat gain to the rooms of not to exceed 0.5 per cent of the average total rate of heat removal for all tests with bare windows. Tests were made to insure that infiltration did not increase above this value with age of the structure, but no account was taken of this small heat gain in the computations.

PRACTICAL APPLICATION OF THE RESULTS

While this report gives laboratory results on windows with southern exposure only, the fact that the results obtained in calculating the heat gain to the rooms check closely with the measured rates of cooling makes it possible to use this method for calculating the heat gain for windows of any other exposure. The factors entering such calculation are the rate of solar radiation, the direction of exposure, and the glass conduction as determined by the inside and outside temperature difference and the wind velocity. The percentage by which this rate of heat gain is reduced by the application of any of the shading appurtenances here reported may be applied to such calculated heat gain through any bare window. Practical values of heat gain through windows and appurtenances under conditions of different exposure and for the weather conditions met with in Pittsburgh during the summer of 1933 are intended to be the subject of another paper.

The average maximum rates and the average of the daily average rates of

[4] Infiltration of Air in Buildings, by O. W. Armspach, A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 121.

heat gain through the windows, given in Table 3 for the conditions studied during the summer of 1933, may be used in design. The average per cent of the solar radiation through a bare window which is transmitted through the same window when a given shading appurtenance is used may be taken from Table 3 as a direct measure of the usefulness of such shading device.

Summary and Conclusions

1. An analysis of the sources of heat gain through a window is made. Calculated heat gains are compared with measured rates of heat removal for bare windows, for completely shaded windows, and for windows with several types of shading appurtenances applied in different ways.

2. Energy transfers through shaded windows are compared with the energy transferred through a bare window. Results showed that 5 per cent of the solar radiation through a bare window was transferred through a completely shaded window, 22 per cent through an outside Venetian blind, 28 per cent through an awning, 45 per cent through an inside shade, and 58 per cent through an inside Venetian blind. Similar percentage transfers are also given for other variations in both kind of finish and application of these appurtenances.

3. Heat removal rates actually required to cool the rooms with the various window appurtenances and the rates of heat removal charged only to the windows are given.

Acknowledgment

The authors acknowledge the valuable assistance rendered by Paul F. McDermott, until recently a member of the Laboratory staff but now associated with the Johns-Manville Research Laboratories.

DISCUSSION

F. C. Houghten: The paper given by Mr. Close and the discussion which followed emphasize the fact that estimation of heating requirements of a building is still a major problem and, as pointed out, is an important project in our research program. Likewise estimation of cooling requirements for summer air conditioning is an important problem and is a project on the Research Committee's program carried out at Pittsburgh, Illinois and elsewhere.

In 1927 the Laboratory at Pittsburgh made a very brief study of the heat impinging, the energy impinging on the surface from solar radiation and the percentage absorption of that energy by the surface, depending upon its character, its color, its temperature and its angle with the sun.

Two years later the Laboratory studied another phase of the problem, namely: the absorption of solar energy by the surface of a roof deck and heat transmission through that deck depending upon its structure.

During the same year J. H. Walker and his associates in Detroit made a study of the cooling requirements of an air conditioned office and showed that the predominating factor in the cooling of such a room was direct solar radiation through the windows.

The results of these various studies pointed the way to the present investigation now being reported, namely: the study of the heat gain of an air conditioned room by solar radiation directly through a window, or resulting transmission, and the effect of curtains, shades, awnings, Venetian blinds and other window appurtenances in reducing that heat gain.

To gain the maximum factor with shading, it is necessary to apply that shading to the outside. When the sun's radiation once penetrates through the glass, it is difficult to remove it, as demonstrated by the former paper. Any type of appliance on the inside is considerably less effective. Shading of any one of several types available is very effective in reducing the cooling requirement.

R. A. MILLER: In connection with the work that Mr. Houghten conducted, we built some small houses on top of our research laboratory and arrived at approximately the same conclusions as Mr. Houghten has reported. During the winter of 1932 to 1933, we found that the heat input necessary to maintain a temperature of 70 F within a double-glazed building was approximately 25 per cent less than the heat input necessary to maintain that temperature in a single-glazed building. That included taking into consideration all the variations in wind velocities and temperature which occurred over a period of approximately three months beginning early in January and ending about April 15.

The temperature differences which occurred when the wind velocities were high, or the outdoor temperatures low, gave a maximum saving of approximately 35 per cent for the double-glazed building as compared to the single-glazed building. The heat input was measured directly by the kilowatt hours meters, the air being circulated quite uniformly throughout the room. We have found subsequently that the velocities used in circulating the air were too high to give the maximum result, and during the current winter we are varying the inside air circulation in order to obtain correlation between what might be called natural circulation and the previous excessive forced circulation. We are planning to carry on this investigation throughout the present winter and the coming summer, using air conditioning as fully as possible, maintaining a relative humidity of approximately 40 per cent in the double-glazed building and a humidity in the single-glazed building just high enough so that frosting will not occur on the glasses. There appears to be a considerable difference in the amount of humidity which can be carried, and we feel this is of considerable importance to future air conditioning problems, and that it will become more and more vital as air conditioning advances.

C. M. ASHLEY: I should like to emphasize one point which I do not believe is entirely clear. It concerns the curve on Page 113 showing the effect of absorption of heat in glasses of different thicknesses. In practice, I believe that due to the greater amount of absorption, the actual transmission of heat to the room will be greater for the thick glass at normal incidence than for the thin glass placed at different angles of incidence. This difference in absorption is due to the fact that the thickness of inclined glass in the direction of travel of the radiation is less than in the direction of incidence.

MR. HOUGHTEN: It probably would be for this reason, that the apparent absorption in a single thickness of glass placed at different angles is not only absorption, as pointed out by one of the speakers, but is partly reflection from the surface, which reflects away that which has completely left the sphere.

There are probably two errors pointed out in the paper in those two curves. The apparent absorption by the glass placed at different angles is partly apparent and some of it is really reflection. Likewise when a thickness by several laminated thicknesses is given, we don't know what happens at those several glass surfaces. Certainly something happens and probably there is a series of partial reflections throwing back energy.

MR. ASHLEY: What compensation has been made for differences of season? Obviously, there will be a difference of angle of incidence of radiation varying with the season, and this will affect the reflection and absorption of the radiation. I wonder how a series of tests run over a period of time can be fully compensated to take this into account.

MR. MILLER: We hope that we will be able to present some of this information

within the next twelve months. The values which we are obtaining will be checked also against the same houses with the windows taken out and blocked off in the same construction as the balance of the houses, and the thicknesses of the glasses will be taken into consideration, as will the seasonal differences of the year.

MR. ASHLEY: Are the percentages of transmission compared with the transmission of clear glass?

MR. HOUGHTEN: No, they are percentage transmission of the total solar radiation impingement on the glass outside surface, and that percentage gain includes transmission gain through the glass as well as radiation.

F. D. MENSING: When I was young—some of the other gray-haired members may remember—it was a practice to build houses with shutters equipped with moving slats at the top and bottom. These shutters were sometimes held partially closed with little tassels. Many of the old houses in Philadelphia have parts of the hinges still imbedded in the wood or brick.

From what we have heard from Mr. Houghten, it is evident that there was a reason in what those old builders did. They had discovered how to add to the comfort of human beings. In the summer those slats were kept at an angle of approximately 45 deg, and in the winter they were closed tight. The old builders may have had something that we have lost track of in later days. We are fundamentally comfort engineers. I think it would be very interesting if Mr. Houghten could find the time to equip his apparatus with the old green shutter with the slats and investigate the results obtained under various conditions.

E. C. EVANS: I want to ask Mr. Houghten if he has verified any experiments on Duplate. I mention that material as a generic term of the double window, non-shatterable type of glass. In considering the use of the two thicknesses of glass in building, it seems to me that there is a great deal to be gained with this type of construction.

MR. MILLER: Duplate is a laminated safety plate glass made up by assembling a piece of plate glass ⅛-in. thick, a piece of plastic material approximately 30-thousandths of an inch thick, and another piece of ⅛-in. thick plate glass. In the case of the product mentioned, the glass was designed particularly to be exclusive of ultra-violet rays and as such becomes exclusive of infra-red. As a transmission medium, it will, of course, cut off just about the same proportion of infra-red and of ultra-violet as solid glass of the same general thickness. As an insulating medium it is equally as effective as plate glass of exactly the same thickness. The transmission value, or rather the insulating value, to be derived from the plastic material between the two glasses is so small as to be virtually immeasurable. We are unable to find any particular value from an insulating standpoint except in the ultra-violet and infra-red exclusion. The use of Duplate as a single glass in a window, if it is double-glazed, is another phase of the problem, but as an insulator, laminated glass should not be considered as having greater value than regular plate glass or window glass of the same overall thickness.

MR. HOUGHTEN: Mr. Ashley's question on the effect of the time of year on the result might warrant further facts. These studies were started in midsummer and carried on into September and, of course, during that time the angle of the sun changed considerably. We based our analysis on the assumption which we thought was somewhat warranted, although somewhat doubtful, that reflection from the surface is a negligible factor. It may or may not be, and if it is not, there will be some errors resulting therefrom.

I might point out also, answering Mr. Evans, that this was not a study of glass; it was a study of window curtains and therefore we could not take time to change glass during the summer season. We decided on one glass and quite a little thought was given to what that glass should be. The glass used was A quality, double-strength window glass.

No. 976

A PROVING HOME FOR AIR CONDITIONING INVESTIGATIONS

By Elliott Harrington * (*MEMBER*), and Leon A. Mears † (*NON-MEMBER*)
Schenectady, N. Y.

THIS is the first of a series of papers to be presented from time to time, on the investigations under actual living conditions at the Proving Home of the General Electric Co. This air conditioned home was unofficially established in September, 1928, for the purpose of testing the oil furnace under actual field conditions. On March 1, 1933, it received official recognition as part of the General Electric Air Conditioning Institute and was set up to accomplish the following four-fold objective:

1. Improvement of air conditioning apparatus, both in production and in development.
2. Improvement and simplification of installation and application standards.
3. Accumulation of additional data on the benefits of air conditioning to comfort and health.
4. General public education on matters pertaining to air conditioning, through the publishing of test reports and through public demonstrations.

General Description

The Proving Home is located in Schenectady, N. Y. This location affords a study of operations under nearly all the climatic conditions met with in this country. Summer and winter extremes of 104 deg above and 24 deg below zero, respectively, have been recorded. The mild spring and fall of this locality round out the picture.

In order to demonstrate that year 'round air conditioning can be applied to the average house, the Proving Home is of ordinary frame construction as indicated by Fig. 1. The only special provisions are the addition of storm sash to all windows, awnings on the sun porch windows for special testing and ½ in. of rigid board insulation on the under side of the roof rafters. Windows are of the usual double-hung construction without weather strip. Doors are also without weather strip and are equipped with combination screen and storm doors. Outside walls are made up of wood shingles, building paper, ship lap, studding, wood lath and plaster.

Fig. 2 shows the plan of the first, second and third floors, as well as the

* Engineer, Air Conditioning Dept., General Electric Co.
† Air Conditioning Dept., General Electric Co.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by Elliott Harrington.

duct layout and location of all registers and radiators. It will be seen that the various types of rooms which go to make up the problem of air conditioning a home, are well represented in this layout:

1. Normal first floor room over heated area (Living Room).
2. Normal second floor room over heated area (Bedroom 1).
3. Normal first floor room over unheated area (Library).
4. Room with large glass exposure over unheated area (Sun Porch).
5. Normal third floor room (Maid's Room).
6. Built-on garage (directly back of Library—not shown).

All rooms of the home are completely furnished in the usual manner, including a fully equipped laundry, recreation room and boys' shop in the basement, as well as the machine room and instrument room.

In spite of the reliable application engineering information available, it is believed that in the present state of the air conditioning art, the data furnished

FIG. 1. THE PROVING HOME IN WINTER

by the various indicating and recording instruments must be supplemented by the actual physical reactions of various people living in the home. Therefore, two elderly people, a middle-aged man, a young man and two children live in the Proving Home at all times.

Table 1 shows the calculated heat gain and heat loss of each room, and Table 2 summarizes these calculations.

DISTRIBUTION SYSTEM

In addition to facilities for installing and testing various types of unit apparatus, two central distribution systems are installed:

1. One pipe steam with basement loop, and wet return.
2. High volume, low pressure duct system.

By means of valves and double throw switches, either system may be used, independently of the other.

The radiator system includes standard height, window height, baseboard and enclosed radiators for study purposes. Vacuum vent valves are used through-

out. The radiator in the sun porch is equipped to operate as a two pipe radiator, in order to study the operation of radiator thermostatic valves.

The duct system is laid out as shown in Fig. 2, and includes the following special features:

1. Four floor returns—one each in the front hall, library, sun porch and third floor stair well.
2. One horizontal delivery ceiling return in the library.
3. Six delivery openings in the library—two in the floor discharging vertically and two each at the baseboard and close to the ceiling, discharging horizontally.
4. Horizontal Ceiling and horizontal floor delivery in Bedrooms 1, 2 and 3.

TABLE 1. HEAT LOSS AND HEAT GAIN CALCULATIONS FOR PROVING HOME

Room	Volume Cu. Ft.	Heat Loss		Heat Gain		Radiation Sq. Ft. E. D. R.	
		Btu per Hour	Heated Air Cfm	Btu per Hour	Cooled Air Cfm	Calculated	Installed[a]
Sun Porch	1020	14080	216	5040	137	58.5	34
Living Room	2450	7820	120	4390	120	32.5	80
Hall	606	2690	41	830	23	11.0	20
Dining Room	1400	4660	71	2930	80	19.5	40
Library	1650	15980	245	4710	128	66.5	62
Lavatory	156	1500	....	Not	Cooled	6.0	5
Breakfast Nook	156	1500	....	Not	Cooled	6.0	5
Kitchen	867	3400	....	Not	Cooled	14.0	15
Entry	260	4710	....	Not	Cooled	19.5	18
Bedroom No. 1	1080	7820	138	3420	107	32.5	20
Bedroom No. 2	886	5850	103	3040	95	24.5	30
Bedroom No. 3	1670	9240	163	3470	109	38.5	40
Blue Room	720	5610	99	2360	74	23.5	38
Bath	394	2130	....	Not	Cooled	9.0	18
Study	600	4110	72	3510	110	17.0	30
Maid's Room	430	3790	78	3010	110	16.0	30
Maid's Bath	250	2610	54	1540	57	11.0	...
Garage	1900	9600	....	Not	Cooled	40.0	40
Totals	16495	107100	1400	38250	1150	445.5	525

[a] This radiation was installed when the house was built in 1924. It indicates how inaccurate Rule-of-Thumb methods can be.

Assumptions

1. *Heating*
   (a) Lowest recorded temperature — 24 F.
   (b) Design temperature — 9 F.
   (c) Indoor temperature 70 F.
   (d) Calculated Relative Humidity, 42 per cent.
   (e) Heated volume, exclusive of garage, 14,595 cfm.
   (f) Degree days in a normal heating season of 8 months, based on 65 F, 6889.
2. *Cooling*
   (a) Outdoor design dry bulb temperature 91 F.
   (b) Outdoor design wet bulb temperature 74 F.
   (c) Indoor dry bulb temperature 78 F.
   (d) Moisture gain 0.17 lb per hour per person.
   (e) Fresh air supply of 300 cfm definitely provided by kitchen exhaust fan. This corresponds to approximately 1¼ changes per hour for entire house.
   (f) Awnings over windows exposed to sun, and attic ventilator are used.
   (g) Cooled volume 12,760 cfm.
   (h) Degree hours in normal cooling season of 4 months based on 12 hours of cooling per day, 7310.
3. *Circulation*
   The cfm requirements for air conditioning were obtained by proportioning the total cfm of the air conditioner according to heat loss or heat gain of room after correcting the Btu requirements to allow for duct losses. 15 per cent was added to Btu of second floor rooms and 35 per cent to third floor rooms. Air conditioner to deliver 1400 cfm of heated air. Same conditioner will deliver 1150 cfm of cooled air.

It will be seen that the library can be conditioned with almost any conceivable type of air flow by using the required combination of supply openings, return openings, delivered air temperatures and air volumes. Ordinary crossbar grilles are used. Ducts are insulated with 4-ply cellular asbestos, which is moisture proofed and sealed. The entire duct system was installed after the house was built, thus showing this procedure to be practical.

The kitchen, garage and bathrooms are heated by steam radiation to avoid recirculation of odors. Also to prevent an undue concentration of cooking

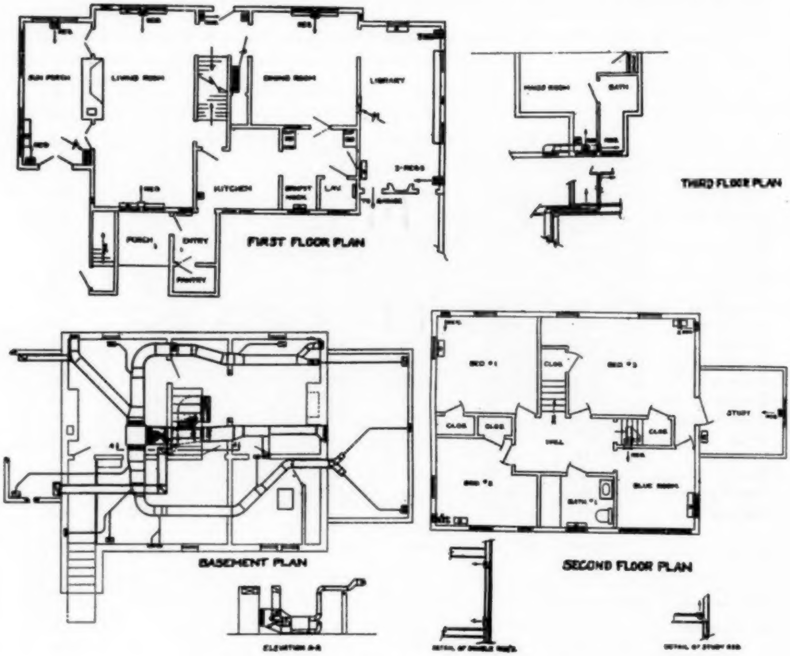


FIG. 2. FLOOR PLAN AND DUCT LAYOUT

odors and to give a measure of temperature control in the summer, the kitchen is forced ventilated to the out-of-doors. Figure 3 shows a typical delivery duct and radiator installation in the library.

MECHANICAL EQUIPMENT

The permanent air conditioning equipment installed in the home consists of:

1. *Oil Furnace*—133,000 Btu/hour maximum output. (Referring to Table 3, it will be seen that the oil furnace is too small for this application, under normal operation. However, the application was made in this manner for the purpose of studying methods of reducing pick-up load, as will be explained in a later report.)

TABLE 2. SUMMARY OF HEAT LOSS AND HEAT GAIN CALCULATIONS FOR PROVING HOME

Heat Loss in Btu Per Hour	
Infiltration	16,000
Conduction through walls	43,900
Conduction through windows and doors	9,000
Conduction through roof	16,000
Conduction through floor over unexcavated basement	4,700
Over-all Exposure Factor 1.09	89,600
Total heat loss above grade	97,500
Basement Loss	13,200
Garage—(50 F)	9,600
TOTAL HEAT LOSS OF HOUSE	120,300
Heat Gain in Btu Per Hour	
Walls	
Conduction due to temperature difference	9,600
Additional conduction due to sun effect	9,000
Windows	
Conduction due to temperature difference	1,900
Direct sun effect	17,800
[b]Roof	
Conduction due to temperature difference	3,100
Additional conduction due to sun effect	10,200
Fresh Air Supply (300 cfm)	12,600
Occupancy (5 persons)	2,000
TOTAL HEAT GAIN WITHOUT ATTIC VENTILATION AND WITHOUT AWNINGS	66,200
TOTAL HEAT GAIN WITH ATTIC VENTILATION AND WITHOUT AWNINGS	56,000
TOTAL HEAT GAIN WITH AWNINGS AND WITHOUT ATTIC VENTILATION	48,400
TOTAL HEAT GAIN WITH ATTIC VENTILATION AND WITH AWNINGS	38,200

[b] NOTE: The conduction through roof due to temperature difference is calculated heat gain through ceiling with an attic temperature of 96 F, and is the gain when an attic ventilator of sufficient capacity to maintain this attic temperature is used. The additional conduction due to sun effect is increase of ceiling conduction due to high attic temperature without attic ventilator.

2. Central air conditioner 130,000 Btu/hour maximum sensible heat output with 1,600 cfm air circulation and a humidifying capacity of 1.5 gal of water per hour (as installed at the Proving Home, the heat output has been reduced to 115,000 Btu/hour with 1400 cfm air circulation).
3. Cooling attachment to the central air conditioner with a heat absorbing capacity of 44,000 Btu/hour and a dehumidifying capacity of 1.5 gal of water per hour.
4. Two refrigerating units, each of 24,000 Btu/hour capacity.
5. Attic ventilating fan of 1,600 cfm capacity.

The oil furnace generates steam between the limits of 0.5 and 6 lb steam pressure to heat the transfer surface in the air conditioner whenever the room thermostat calls for heat. The oil furnace also heats domestic hot water the year 'round, by means of an indirect heater. Further, it is used to supply steam for the radiation heating system.

The air conditioner draws air through the return system, over the cooling and dehumidifying surface and through the metal wool filters, delivering the air over the heating and humidifying surfaces to the delivery duct system.

The refrigerating units operate on a call from the room thermostat for cooling, to cool the extended surfaces at the return entry to the air conditioner. Either direct expansion or indirect cooling is available for testing purposes.

Fig. 4 is a general view of the machine room, showing the oil furnace and refrigerating units. Fig. 5 shows the air conditioner with cooling attachment.

FIG. 3. DELIVERY DUCT AND RADIATOR INSTALLATION IN THE LIBRARY, SHOWING THERMOCOUPLES TO MEASURE AIR INLET TEMPERATURES

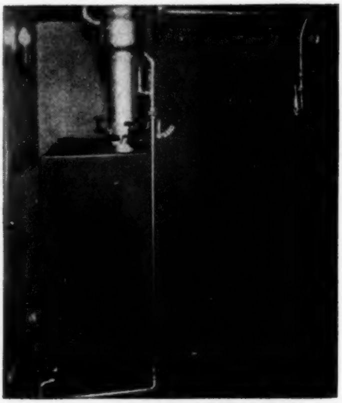


FIG. 5. CENTRAL AIR CONDITIONER WITH COOLING ATTACHMENT

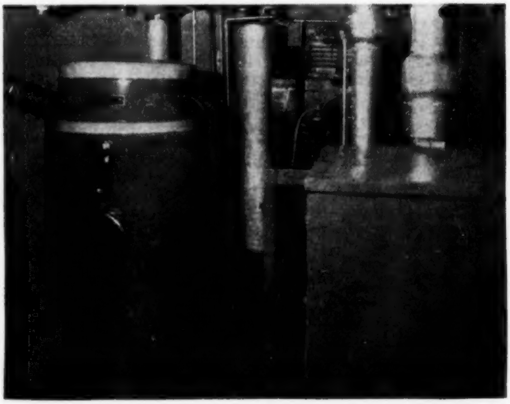


FIG. 4. MACHINE ROOM, WITH OIL FURNACE AND REFRIGERATING UNITS

FIG. 6. AWNING INSTALLATION ON SUN PORCH

The attic ventilating fan is arranged to take air in at one end of the attic and expel it at the other end, whenever the attic temperature is higher than the temperature out-of-doors. It is installed so it can also be used during the night to draw cool night air up through the house and out through the attic.

Control System

The control system is made up as follows:

1. An electric driven time thermostat located in the dining room operates to control the starting and stopping of the oil furnace as required for temperature control, when using the radiation heating system. (To date, no humidity control has been provided with the radiation heating.) This same thermostat controls winter temperatures with the air conditioning system by starting and stopping the blower and oil furnace as required. It also controls summer temperatures by starting and stopping one or both of the refrigerating units, depending upon the cooling load. The capacity of the refrigerating units is such that the maximum dry bulb cooling which can be obtained is about 15 F.
2. Humidifying is controlled by a humidistat located in the dining room and operating a solenoid valve in the water supply line to the humidifier. Dehumidification is inherently controlled by the predetermined temperature of the cooling surface.
3. The radiation in the kitchen and bathrooms is controlled by an independent room thermostat operating a motor-valve in the steam supply line, regardless of whether the radiation or conditioning system is being used.
4. The garage radiator is controlled by another independent room thermostat and motor valve combination, so interlocked that the valve can admit steam to the radiator only when the garage doors are closed.

Measuring Equipment

Temperatures are measured by 102 thermocouples mounted at various levels and in various locations in each room as well as in the attic spaces, basement spaces, out-of-doors, and in certain parts of the machine equipment and duct system. These thermocouples are read on a potentiometer through a selector switch system and also may be automatically recorded on a recording potentiometer. Temperatures are also continuously read and recorded on three-day or seven-day charts, in the dining room, library, sun porch, garage, kitchen, bedroom No. 1 and maid's room, as well as out-of-doors.

Humidity is recorded continuously both inside and out, the recording instruments being regularly checked with a sling psychrometer. Humidity distribution is so even that it is felt unnecessary to record it at more than one place.

Air velocities are measured by sensitive electric hot wire anemometers. All suspensoids collected by the filters are analyzed for composition and for size of particles. A dust counter is used to determine the dust content of the air both inside and out.

Operating time is totalized for each operating machine, using individual electric time meters. In addition, on-off operating cycles are recorded on a chart, for each operating machine.

Electric power consumption is totalized for each machine on individual integrating watthour meters.

Oil consumption is read periodically from a calibrated direct reading meter.

Water consumption is totalized for each machine on integrating meters.

Wind velocities are obtained from the weather bureau.

All instruments are calibrated by and under the supervision of the General Engineering Laboratory.

SPECIAL EQUIPMENT

1. The sun porch is equipped with awnings and this room is used particularly to study heat gain calculations. (Fig. 6.)

2. A cooling tower is available and it is planned to conduct a series of

TABLE 3. BOILER LOAD FOR PROVING HOME

Air Conditioning		Steam Heating	
[c] Air Conditioner Rating	115,000	Direct Radiation in House (485 sq. ft.)	116,400
[c] Direct Radiation in Spaces not Air Conditioned (61 sq. ft.)	14,650	Direct Radiation in Garage (40 sq. ft.)	9,600
[c] Direct Radiation in Garage (40 sq. ft.)	9,600	Pipe Tax (14 per cent of 525 sq. ft.)	17,600
[d] Pipe Tax (14 per cent of 101 sq. ft.)	3,400	Pick Up (25 per cent) of 525 sq. ft.)	31,500
[e] Pick Up	.....	[f] Domestic Hot Water	.....
Domestic Hot Water	12,500		
Total Boiler Load...155,150 Btu/hour		Total Boiler Load...175,100 Btu/hour	

NOTES:
[c] The air conditioner *rating* for the blower speed employed is used here; not the heat loss of the air conditioned space, since the former is the actual load imposed on the boiler during periods of operation. For the same reason, *installed* radiation is used, and not calculated.
[d] One pipe steam system with all mains, insulated.
[e] No pick-up allowance for split system since direct radiation is only 21 per cent of air conditioner rating.
[f] Domestic hot water neglected in steam system since it is only 40 per cent of pick-up alllowance.

tests during the summer of 1934, to determine the possibilities of such a device for condenser water cooling, in homes.

3. A recording ion counter has been developed by the Research Laboratory and is installed in the library of the Proving Home. A study of the ion count with various degrees of habitation and various climatic conditions, is being carried out.

TEST RESULTS

An intensive schedule of investigation work is under way, covering a period of at least two years. As stated at the beginning of this paper, it is planned to present certain of the data obtained in the Proving Home, at regular intervals. Thirty-one separate investigations are under way or have been completed to date. The results of four of these investigations are presented here with the hope that they may be of interest and help to the air conditioning industry.

I. GENERAL WINTER OPERATION TEST

From November 15 to December 13, 1933, the air conditioning system at the Proving Home was operated normally by the regular automatic controls. An exception to the above was taken during the four day interval when the air conditioning system was shut down and the house heated entirely by steam radiation. This latter was done to provide a comparison between steam radiator heating and complete air conditioning. In order to insure a true

TABLE 4. NORMAL WINTER RUN

(Nov. 15—Dec. 13, 1933)

Type of Heating, Etc.	Dates of Test	Time in Hours	Oil Consumption (Gallons)	Power Consumption (Kwhr)	Degree Days (65 F. Base)	Oil Cost Per Normal Season (of 6889 Deg. Days)	Power Cost Per Normal Season (of 6889 Days)	Total Operating Cost per Normal Season (of 6889 Deg. Days)
Steam	(11-30-33) to (12-4-33)	93.75	41	7.8	123.3	$137.90	$13.00	$150.90
Complete Air Conditioning	(11-15-33) to (11-30-33)	354.75	200	64.5	505.8	163.50	26.40	189.90
Complete Air Conditioning	(12-4-33) to (12-13-33)	217.0	143	46.7	369.8	159.90	26.10	186.00

Oil Furnace Total Operating Time (Minutes)	Air Conditioner Total Operating Time (Minutes)	Operating Time of Air Conditioner in Per Cent of Total Duration of Test	Operating Time of Oil Furnace in Per Cent of Total Duration of Test
2004	....	..	35.7
9438	8525	40	44.4
6980	6261	48	53.6

NOTE: Degree days figured from Proving Home temperature charts, and based on average hourly temperatures.

picture of the operation, the engineer supervising the tests did not manipulate the apparatus or automatic controls in any way. His entire time was spent in recording all possible data. Following is an outline of the records kept:

1. Temperature on all regular indoor and outdoor recording thermometers.
2. Humidity recorded indoors (1st floor), outdoors, and in the garage; with sling psychrometer readings taken on the second floor and on the first floor to check the recording instruments.
3. Operating cycles recorded for the Oil Furnace and Air Conditioner.
4. Power consumption of all equipment.
5. Daily domestic hot water consumption.
6. Total operating time of all equipment.
7. Oil consumption.
8. Dust accumulation in living quarters with dust count readings taken at intervals, both indoors and out.
9. Condensation conditions on all windows.
10. Temperatures between storm sash and inside windows.
11. Quantity of hot water stored in domestic hot water tank at frequent intervals.
12. Outside weather conditions.
13. Special tests. Some of these will be summarized later in this report.

Table 4 gives a brief summary of the operation.

The operating costs were calculated on the basis of electric power costs at 3c per kwh and No. 4 oil at 6c per gallon. The degree days were figured on a

65 F basis. In order to obtain the cost of operation for a normal heating season, the cost for the period of the test was multiplied by the ratio of the degree days in an average Schenectady heating season (6889) to the degree days during the test.

The heating costs for steam are slightly lower than they should be. As the oil furnace is purposely overloaded, about 30 per cent, two or three of the radiators at the end of the one pipe steam system were never entirely hot and the heating of these rooms suffered proportionately. A previous check at the Proving Home, over a complete heating season and with a similar type of steam heating equipment of adequate capacity, showed a total heating cost for a normal season of $159.00. The overload on the furnace is not so apparent,

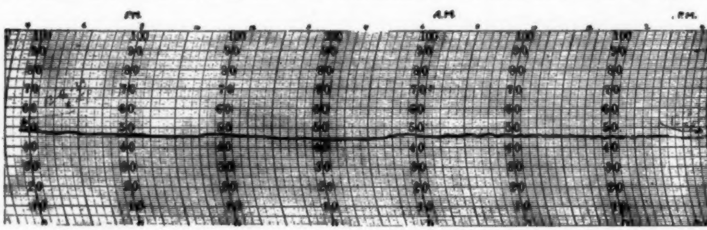


FIG. 7. RELATIVE HUMIDITY CHART FOR ONE DAY DURING NORMAL WINTER RUN

when using the air conditioning system, because of even heat distribution and, as a result, these air conditioning costs are reasonably accurate.

A 60 deg temperature was maintained at night, which accounts for the low percentage of the total *time-on* for the air conditioner.

During the test the air conditioner filters collected about one ounce of dust. This dust collection is less than would be expected. It can be accounted for by the fact that the dust count in the Proving Home had been reduced to below 30 particles per cubic centimeter, with the out-of-door count averaging less than 50 particles per cubic centimeter during the test. The collected dust was very fine, some of it being but one micron in diameter.

The following outline indicates the advantages and disadvantages of the complete winter air conditioning over heating the house by radiation, as shown by this test.

Advantages

1. Substantial reduction of dust accumulation in the living quarters. (The maid found it necessary to dust only every third day instead of every day, as with radiation heating.)
2. Humidification under positive control. (See Fig. 7 for the record of humidity control for a typical day at the Proving Home during the test.)
3. Closer temperature control. (See Fig. 9 for temperature record in Dining Room with steam heating and with air conditioning.)
4. Uniform temperature more readily obtained throughout house. (See Table 5, for typical temperatures taken almost simultaneously. See Figures 8 and 10 for 3-day temperature records in Dining Room, Library, and Second Floor Bedroom.)
5. Floor to ceiling temperature differential decreased. (See Test No. III.)
6. Elimination of normal tobacco smoke odors. This occurs since the smoke is diluted and distributed through the house and its absorption into walls and solid

materials is increased by the greater area available and by the flocculation accomplished by the air circulation.

Disadvantages

1. Operating cost approximately 15 per cent greater than for steam heating. (This is not a general conclusion. *For equal comfort,* the operating cost of the air conditioning system should not exceed the operating cost of the radiation system by more than the extra power required for air circulation. The excess cost may be less than this due to lower mean temperatures for the same degree of comfort.)
2. Higher first cost in remodeled heating systems.

TABLE 5. BREATHING LINE TEMPERATURES OF INDIVIDUAL ROOMS IN DEGREES FAHR WITH COMPLETE AIR CONDITIONING SYSTEM IN OPERATION, AFTER REACHING STEADY STATE CONDITIONS

3rd Floor Bath	75	2nd Floor Study	74
3rd Floor Bedroom	74	1st Floor Hall	74
2nd Floor Bedroom No. 1	73	1st Floor Living Room	75
2nd Floor Bedroom No. 2	73½	1st Floor Library	78½
2nd Floor Bedroom No. 3	74	1st Floor Dining Room	74
2nd Floor Hall	74	1st Floor Sun Porch	75½
2nd Floor Blue Room	74		

1st floor library maintained at higher temperature because of cold floor conditions. (Room over un-excavated space.)
Controlling thermostat set at 74 F.
Temperatures measured at center of room, at the breathing line.

TABLE 6 — DOMESTIC HOT WATER CONSUMPTION

Duration of Test (Hours)	665.5
Total Hot Water Used (Gallons)	1,795
Average Daily Consumption (Gallons)	64.7
Maximum Daily Consumption (Gallons)	105.5
Minimum Daily Consumption (Gallons)	30.5
Temperature of Hot Water (F)	125
Average Wash Day Consumption (Gallons)	95.6
Average Daily Consumption Excluding Wash Day	60.6

TABLE 7. MAXIMUM AND MINIMUM FLOOR TO CEILING TEMPERATURE DIFFERENTIALS WITH STEAM HEATING AND WITH AIR CONDITIONING UNDER NORMAL OPERATING CONDITIONS

Room	Air Conditioning								Steam Heating		
	Differentials		Air Flow Cfm	Out-side Temp. F.	Differentials		Air Flow Cfm	Out-side Temp F.	Differentials		Out-side Temp. F.
	Max. F.	Min. F.			Max. F.	Min. F.			Max. F.	Min. F.	
Sun Porch (unheated floor)	11¼	8	150	24 degrees to 12 degrees	7	3½	150	46 degrees to 44 degrees	14	11¾	34 degrees to 28 degrees
Living Room	6	3¼	65		3½	1¾	65		8¼	5¼	
Bedroom No. 3	11	7	80		9	4½	80		14½	8¾	
Library (unheated floor)	15½	8	140		9½	3½	140		17½	9¾	

(This table does not include differentials during long morning pick-up heating cycle.)
Temperature of air supplied to rooms varies between 90 and 135 F.

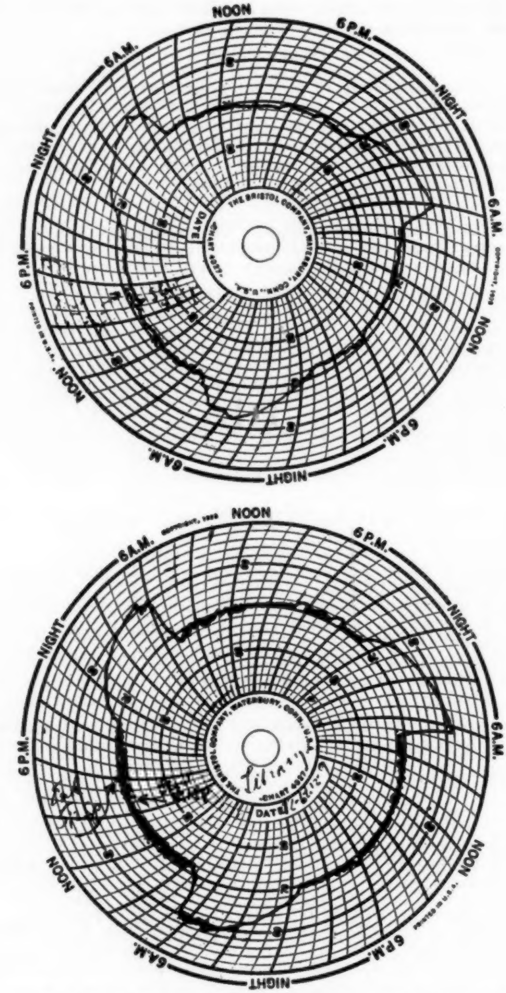


FIG. 8. TYPICAL DRY-BULB TEMPERATURE CHARTS FROM LIBRARY AND BEDROOM NO. 3, SHOWING TEMPERATURE BALANCE IN VARIOUS PARTS OF THE HOME WITH AIR CONDITIONING. CONTROLLING THERMOSTAT IN DINING ROOM SET TO HOLD 71 F.

II. DOMESTIC HOT WATER CONSUMPTION

During the normal winter tests, daily hot water consumption was recorded. (See Figs. 11 and 12.) The occupancy of the house during this period was

three adults, one child, and one maid. The household was not informed of the recording in order to keep the water consumption absolutely normal. The 24-hour periods are from 4:00 p. m. of one day to 4:00 p. m. of the next. This time was chosen as it was inconvenient to record from midnight to midnight. Table 6 summarizes the daily hot water consumption records.

The outlet of the hot water storage tank is connected through a thermostatic mixing valve, so that the outlet water temperature is held constant at 125 F plus or minus about 2 F.

III. Temperature Differential Between Floor and Ceiling

In recording floor to ceiling temperature differentials (hereafter referred to as differential), the air was supplied to the rooms at temperatures varying from 90 to 135 deg. The following factors were found to affect the differential and with each test, mention will be made as to how certain of these factors were controlled.

1. Blower operating time, prior to reading temperatures.
2. Temperature of air delivered to room.
3. Outside temperature.
4. Rate of air delivery.
5. Location of supply and return grilles.
6. Type of heating system.
7. Construction of room.

It was found that if air of a constant temperature and of sufficient volume to provide about six air changes per hour was delivered to a room, the differential became practically constant in 2 to 3 min.

Further, it was found necessary to be able to measure the temperatures with thermocouples. Temperature changes were effected within a few seconds and as a result it was possible to get true readings, only with a thermocouple of small mass, which followed the air changes closely. All differentials were read at the center of the room.

Test "A"

Maximum and minimum differentials were first obtained under normal operation with steam heating and with air conditioning. Table 7 summarizes the most pertinent data obtained.

In this test, readings were taken every half hour for 8 hours during the day, but did not include the long morning pick-up heating cycle. The readings were taken without setting up special conditions, so they cannot be compared too closely. However, considering the number of readings, it is believed that typical operating results were obtained. The lower horizontal supply and return grilles in the library and the lower horizontal supplies on the second floor were used. Other supplies were all vertical from the floor. Readings were taken for the air conditioning system, whether the blower was operating or not, but a sufficient number of readings were taken with the blower operating to get representative differential readings. Outside temperature conditions varied throughout but are given in Table 7, with the other tabulations.

Test "B"

Maximum differentials with various combinations of supply and return grilles in the library, were studied.

TABLE 8. MAXIMUM FLOOR TO CEILING AND BREATHING LINE TO CEILING TEMPERATURE DIFFERENTIALS IN LIBRARY WITH DIFFERENT COMBINATIONS OF SUPPLY AND RETURN GRILLES

(After Stable Condition Reached for Given Supply Air Temperature)

Combinations of Supply and Return Grilles Used[a]	Maximum Floor to Ceiling Differential F.	Maximum Breathing Line to Ceiling Differential F.	Cfm Air Delivery	Outside Temp. F.	Supply Air Temp. to Room F.
1	11.5	1.9	240	26	113
2	25.5	11.5	240	28	118.5
3	24.9	12.9	240	28	114
4	12.5	1.8	240	29	123
4	14.5	2.0	140	24	120
5	26.5	7.2	240	29	122
Steam Heat	21.5	4.0	...	38	.....

[a] Key to grille combinations:
1. Lower Horizontal Supply—Upper and Lower Horizontal Return.
2. Upper Horizontal Supply—Upper and Lower Horizontal Return.
3. Upper Horizontal Supply—Upper Horizontal Return.
4. Lower Horizontal Supply—Lower Horizontal Return.
5. Upper Horizontal Supply—Lower Horizontal Return.
6. Lower Horizontal Supply—Upper Horizontal Return.
7. Lower Vertical Supply—Lower Horizontal Return.
8. Lower Vertical Supply—Upper Horizontal Return.
9. Lower Vertical Supply—Upper and Lower Horizontal Return.

In this test the outside temperature remained practically constant. The delivery air temperatures varied slightly and enough to need correction. The rate of air delivery was held constant and the supply air temperature was held constant long enough to produce a steady state condition. Table 8 gives the summary with the duct combinations referred to.

Test "C"

Differentials with various combinations of supply and return grilles in the library, taking readings 10 min after the blower started operating, were obtained.

In this test the blower started to deliver air at about 90 F and the temperature gradually rose during this test. (This is typical of one cycle of the air

TABLE 9. FLOOR TO CEILING AND BREATHING LINE TO CEILING TEMPERATURE DIFFERENTIALS IN LIBRARY, WITH DIFFERENT COMBINATIONS OF SUPPLY AND RETURN GRILLES

Combinations of Supply and Return Grilles Used	Floor to Ceiling Differential F.	Breathing Line to Ceiling Differential F.	Cfm Air Delivery	Outside Temp. F.	Supply Air Temp. at Time of Reading F.
1	9½	3½	140	33	104
3	12½	10	140	31	95
4	8½	2¼	140	30	100
5	14	7	140	33	98
6	8½	3½	140	32	113
7	10¼	2	140	34	102
8	12½	2¼	140	32	113
9	11½	2	140	32	101

NOTE: For key to grille combinations, see Table 8.

conditioner operation.) The air delivery, outside temperature and time of taking readings were constant throughout. See Table 9, for details.

Test "D"

Differentials with steam heating and with air conditioning after a prolonged heating cycle and a constant differential had been reached, were obtained.

This test was taken during the morning heating-up cycle, so that a long period of heating could be obtained. It will be noticed that the outside

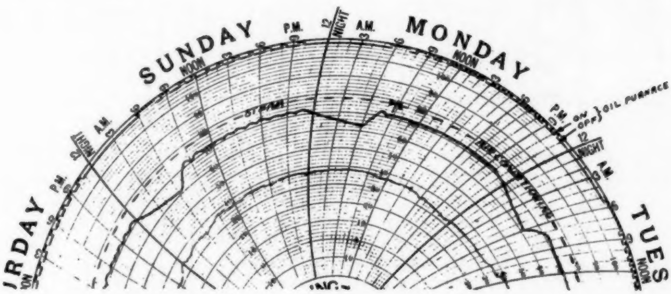


FIG. 9. COMPARISON OF DINING ROOM TEMPERATURE MAINTAINED WITH STEAM HEATING AND WITH AIR CONDITIONING. HEAVY LINE SHOWS DINING ROOM TEMPERATURE. LIGHT LINE SHOWS OUTSIDE TEMPERATURE

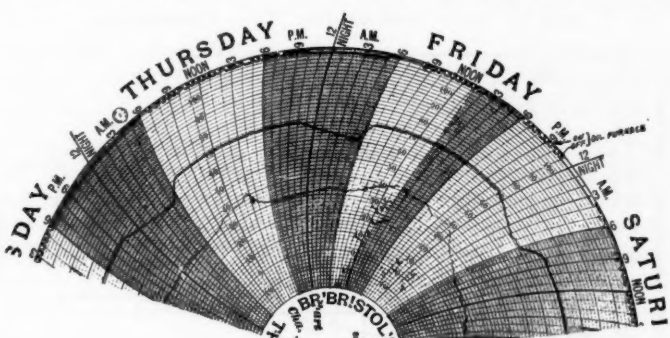


FIG. 10. DINING ROOM TEMPERATURE WITH AIR CONDITIONING AND WIDELY VARYING OUT-OF-DOOR TEMPERATURE. SAME PERIOD OF TIME AS SHOWN IN FIG. 8. HEAVY LINE SHOWS DINING ROOM TEMPERATURE. LIGHT LINE SHOWS OUTSIDE TEMPERATURE

temperature was much higher for the steam heating test than for the Air Conditioning Test, which factor indicates that the contrast between the two cases would be more pronounced than the figures indicated. In the library the lower horizontal supply grille was used while in the other rooms a lower vertical supply was made use of. Table 10 gives the tabulation.

Conclusions

1. The air conditioning system reduced the differential of steam heating from 25 to 50 per cent.

2. The lower horizontal supply with a lower horizontal return gives the

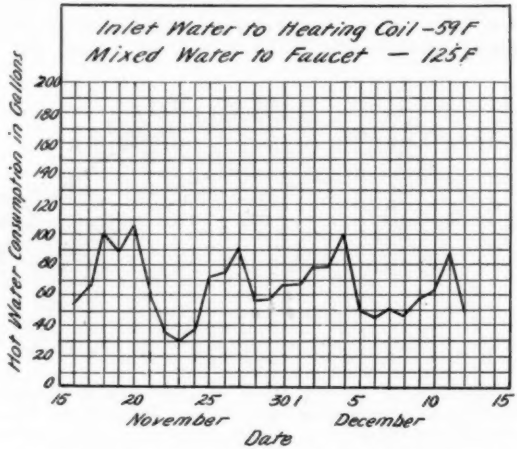


FIG. 11. DAILY DOMESTIC HOT WATER CONSUMPTION

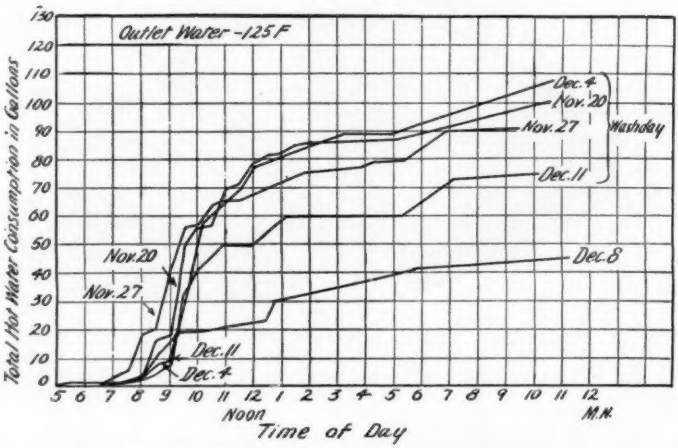


FIG. 12. DOMESTIC HOT WATER CONSUMPTION

lowest differential, with a given air delivery temperature and volume, for winter air conditioning.

Table 10. Floor to Ceiling Temperature Differentials after a Prolonged Heating Cycle and a Constant Differential Had Been Obtained

	Sun Porch	Library	Living Room
Steam Differential, F	12	21½	9
Outside Temperature, F	38	38	38
Air Conditioning Differential	11	14½	5
Outside Temperature, F	24	24	24
Supply Air Temperature, F	125	120	125
Cfm Air Delivery	150	140	65

3. The living room presents an ideal condition for low differentials with a warm floor, due to heated space directly underneath.

4. The library with cold floors, due to no excavation underneath, presents the worst condition met with and future work will be carried on as to the best methods of remedying such a condition.

IV. Summer Air Conditioning

The summer air conditioning equipment was not completely installed in the Proving Home until late in the summer. As a result, most of the time during warm weather was spent in testing the apparatus itself. However, with the apparatus ready for operation, a comprehensive program has been outlined for the summer of 1934.

From the limited data obtained, it was possible to calculate the power cost for an average Schenectady summer season of 4 months and 7310 degree hours, based on 78 F indoor dry bulb temperature. This figure was $116 based on 3c power, but it is subject to further checking before acceptance.

From observations made on ceiling to floor temperature differential in the summer, the maximum differential recorded was 5 deg, based on horizontal delivery just below the ceiling, and floor return. Consequently, the floor to ceiling differential probably does not hold the importance in summer air conditioning that it does in winter air conditioning.

DISCUSSION

A. P. Kratz: Mr. Harrington made the statement that he considered it desirable to have a temperature differential between floor and ceiling of approximately ½ deg per foot, and that it was expected that it would be possible to obtain this difference. I think it is important to state the outdoor temperature at which he expects to obtain it.

We find, in our study of warm-air systems in particular, that in mild weather, possibly between 40 and 50 deg outdoors, we have no difficulty in obtaining a temperature differential between floor and ceiling as low as 3 deg, which with a 9-ft ceiling would be about ⅓ deg per foot.

On the other hand, with 0 deg outdoors, under exactly the same conditions, the temperature differential will be approximately 15 deg from floor to ceiling.

ELLIOTT HARRINGTON: Answering Professor Kratz' question, I think we can obtain the differential of approximately ½ deg per foot, at 0 deg outside. The problem is to do it with unheated space below the floor. We have already obtained it in the living room, as you will note by the data included in the paper. We are running between 2 and 4 deg with 0 deg outside. The conditions there are favorable because it is over the machine room and the floor runs at a temperature of a little more than 70 F. The problem is to do it in a room with the floor located over unheated space. I believe that even in such a location, a differential of ½ deg per foot can be obtained with proper forced air circulation.

No. 977

LOW-COST AIR CONDITIONING FOR A SMALL RESIDENCE

By M. K. Drewry * (*NON-MEMBER*), Milwaukee, Wis.

THE modification of a warm-air system for hot-water air heating, tap-water cooling, automatic fuel feed, and year-round air filtration and humidity control has afforded the experiences and data recorded herein. The eight points of greatest interest are: (1) 90 per cent average boiler efficiency, (2) $6 per summer tap-water cooling, (3) year-round coke fire, (4) 8.5 unit heat transfer rate from a 167-sq ft automotive-type radiator of 1 cu ft volume but adequate for heating and cooling 17,000 cu ft space, (5) 10 per cent CO_2 average from coke fire, (6) 160 F average flue gas temperature, (7) 70 watts input for supplementary mechanical air circulation, and (8) satisfactory stoking by gravity of small-size fuel from a simple welded hopper. Total investment and annual operating costs, including cooling, were found to be no greater than those of the previous simple heating system.

The Installation

Figs. 1 and 2 show the equipment which was used. On February 18, 1933, it replaced a standard warm-air furnace in a 17,000-cu ft residence at Milwaukee, Wis., utilizing the same air circulating system as did the original furnace (Figs. 3, 4, and 5). That this heating system is a combination warm-air and hot-water system, employing water as a heat-transfer medium between the fuel-combustion process and the air-heating procedure, is apparent. A small hot-water boiler (18 in. firepot) provides heated water to an automotive-type radiator mounted in a separate casing through which the room-air circulates. Fuel is fed into the boiler by gravity from a welded sheet metal hopper.

For moderate rates of heat output to the rooms, the air flow is induced by natural convection. When a greater heat output is required, either in cold weather or during rapid room heating, the propeller-type fan comes into service. The fan starts automatically when the water temperature at the top of the boiler approaches the boiling point.

A thermostat in the rooms regulates the heat output and a damper regulator located on the boiler serves to start and stop the fan and to limit the heat output of the fire from exceeding the capacity of the radiator. Thus, operation of the unit is automatic, except for filling the fuel hopper and shaking ashes once daily for the average heating demand per day.

* Asst. Chief Engr. of Power Plants, The Milwaukee Electric Railway & Light Co.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by G. L. Larson.

Summer operation for cooling consists of admitting tap-water to the heat-exchanger and operating the fan continuously to supply cooled air to each room through the same riser pipes and the registers as are employed for heating. The hot-water boiler is isolated from the radiator throughout the summer by closing gate valves, and is then employed only to heat service water. Because of the low heating requirements of service water, fuel additions are made only once weekly during the summer.

DESIGN

To accomplish the main object of providing acceptable year-round automatic air conditioning without exceeding warm-air furnace costs required consideration of many combinations of equipment. Application of rapidly-advancing

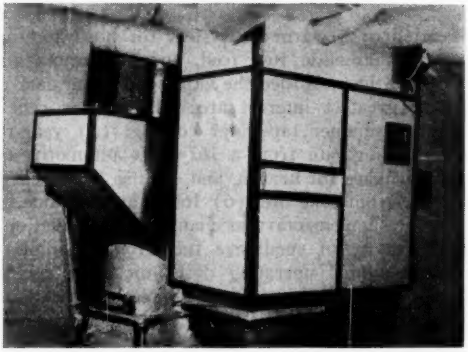


FIG. 1. THE YEAR-ROUND AIR CONDITIONING UNIT INSTALLED IN 1933

power plant practices and principles assisted materially. Making dual or treble use of apparatus, working each piece of equipment to maximum capacity and efficiency, and utilizing standard products as much as possible, helped to attain the desired result. Automotive-type radiation, because of its extremely high heat-transfer efficiency, low cost, small space requirements, and simple pipe connections, could be used because low water pressure could be employed and deterioration prevented by attention to boiler-water conditioning. High reliability and adequate free air-flow area past its blades when idle prompted choice of a propeller-type fan.

Hopper fuel supply permitted choice of a small boiler with adequate small grate area for high efficiency at average ratings, yet ability to carry peaks, and with ability to maintain extremely low heat outputs satisfactorily for mild weather space heating and for service-water heating in summer. Flue gas temperatures were maintained low without additional expense by utilizing the smoke pipe as an air heater by making it air tight and locating the check door in the chimney. The temperature drop in the smoke pipe equals 300 deg at maximum output.

Propeller fan and motor tests, combined with calculations of piping resistances

deduced from circulating air temperatures with the warm-air furnace operating, disclosed that adequate circulation could be obtained with a 1/20 hp motor using only 70 watts. The size of the humidity pan was calculated from a simple test of evaporation rate from an open, heated dish. Table 7 summarizes the principal design data for all of the major apparatus.

Preliminary Tests

Early in February the hot-water boiler, casing, and radiator of the new installation were assembled near the furnace, a smoke pipe connection was

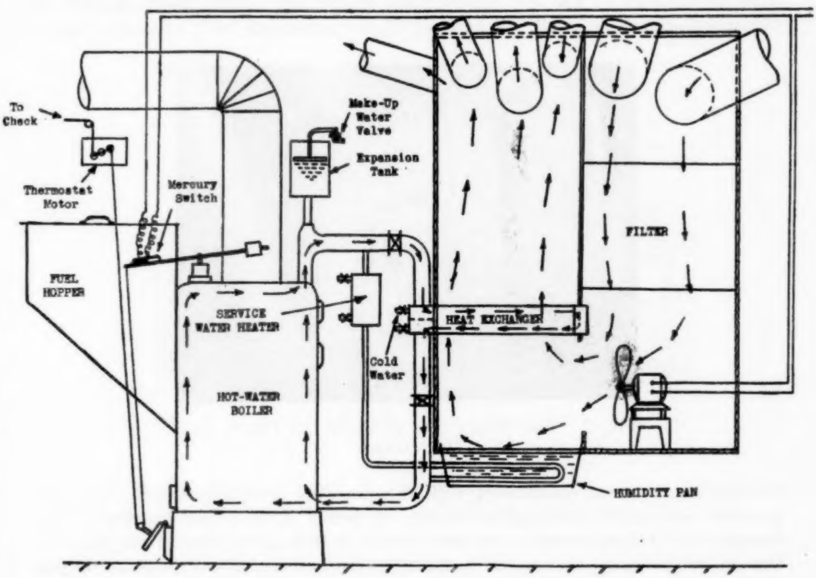


Fig. 2. Air Conditioner Diagram Showing Combination Hot-Water and Warm-Air System for Winter and Summer Service

provided to the chimney, and preliminary tests were made. Data showed that the unit would furnish 90,000 Btu per hour, which was the desired output. On one occasion it was brought from cold, starting the fire with kindling wood, to 90,000 Btu per hour, in only 23 min. This experience proved the value of the small water storage (160 lb) in the system. Operating experiences, with the unit serving for typical residence duty during the subsequent heating months and cooling months, were observed and recorded in detail as material for an engineering thesis [1] on which this paper is based. Enough data were collected to approximate efficiency and capacity reliably each day for two heating months. More detailed data were taken during three shorter tests to learn the characteristics over a wide range of ratings. All test procedure followed accepted methods as far as conditions permitted. Air flow was

[1] University of Wisconsin, 1933.

measured with a calibrated anemometer and the thermometers were checked. Tables 1 and 2 summarize the daily data collected during March and April and Fig. 7 shows room and heated-air temperatures on the coldest day experienced. Fig. 8 shows boiler test data when thermostatically controlled to average about 25 per cent output.

OPERATING EXPERIENCES

Room Temperature Regulation

A lesser vertical temperature gradient in the rooms was readily noticeable after installation of the new system; and temperatures were more uniform in horizontal planes, as well. Where 75 F, as registered by a centrally-located

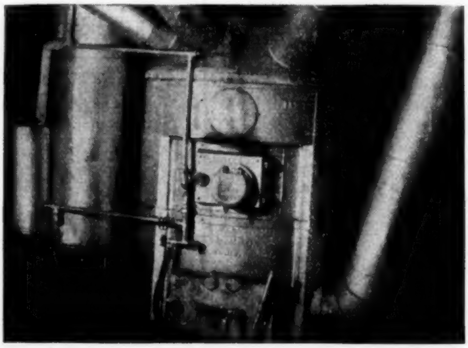


FIG. 3. THE WARM-AIR FURNACE AS ORIGINALLY INSTALLED

thermometer, caused discomfort with the previous installation, the same temperature was nearly acceptable because of better vertical and horizontal uniformity of air temperature. Another proof of this uniformity is the fact that the centrally-located thermostat in an unheated hall consistently checks the fire within two or three minutes after the fan starts. Many charts were secured showing no more than 2 F maximum variation of average room temperature. (See Fig. 7.) Air issues from the wall registers at about a 45 deg angle when the fan operates, instead of rising almost vertically when induced by natural convection as was the case with the original furnace. The velocity is not objectionable, rather it is desirable for inducing liberal air movement.

Fan Noise

Though the fan and motor were quieter than average portable room ventilating fans when tested outside of the casing, their noise was not acceptable when confined in the casing and released through the tin leader pipes into the rooms. Wall-board had been used for the casing to diminish the sounds, but not until the casing was lined with 3/8 in. of hair felt was the problem acceptably solved. The sound can now be likened to that of a quiet oil burner in a warm-air furnace. Since it is of relatively low frequency, its total elimination by absorption is difficult.

Ashes and Clinkering

Experience showed that *slicing* the ash bed, by means of a ⅜-in. rod inserted through a hole in the clinker door, reduced the combustible in the ash from 8 per cent to 5 per cent and improved the CO_2. If this was done before shaking, and shaking discontinued when bright coals appeared in the ash pit, downward ash flow was continuous and the firebed was kept in a stable condition.

A second type of coke clinkered more readily than the first kind, though all clinkers were small and were readily broken by slicing to pass through the grates without any trouble. After a capacity test of 10 hours at continuous maximum rating with the less favorable coke, clinkering was no different than normal. The same coke, when hand fired, often formed large clinkers

Fig. 4. The Milwaukee Residence Which Is Air Conditioned

that filled the entire firepot. It appears that the continuous gravity feed eliminates localized high temperatures attendant on periodic hand firing.

Ashes removed checked very closely with weights calculated from the coke analyses. For the first ton, the disparity was only 3 per cent. (See Table 3.)

CO Experiences

CO was first detected in the flue gas after the coke hopper nearly emptied on one occasion and considerable green coke was put onto the fire when re-charging the hopper. The rapid evolution of CO forced flue gas from a leak in the smoke pipe, and long bluish-white flames were observed through the cleanout doors, passing entirely through the boiler into the smoke pipe. Orsat readings taken at 5-min intervals showed about 16 per cent CO_2 and the following percentages of CO: 5, 3, 2.6 and 3.4. In 30 min about one per cent CO was averaged. Twelve hours later some was still detected, and the average for the 24 hours was 0.4 per cent.

This experience proved that the phenomenon of CO_2 reduction to CO occurs in a thick incandescent fuel bed. Unless air is supplied over the fire, appreciable losses occur, since one per cent CO represents about 5 per cent fuel loss. To supply over-fire air continuously is an undesirable corrective, for it can readily cause greater sensible heat loss in the flue gases than it saves in

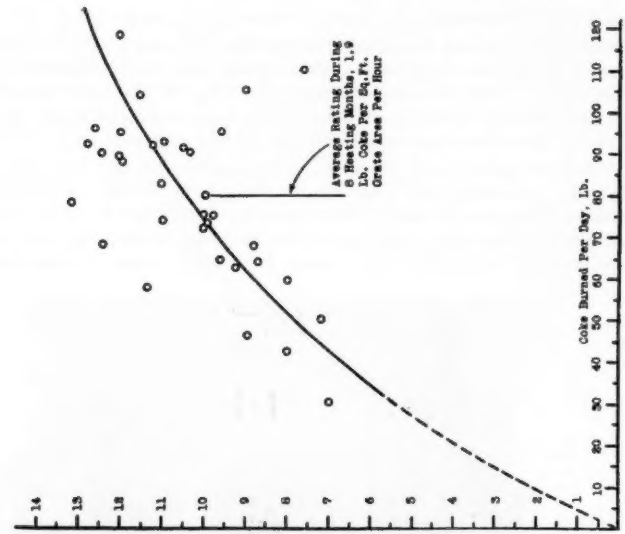


FIG. 6. CO_2 IN FLUE GAS VERSUS COKE BURNED PER DAY

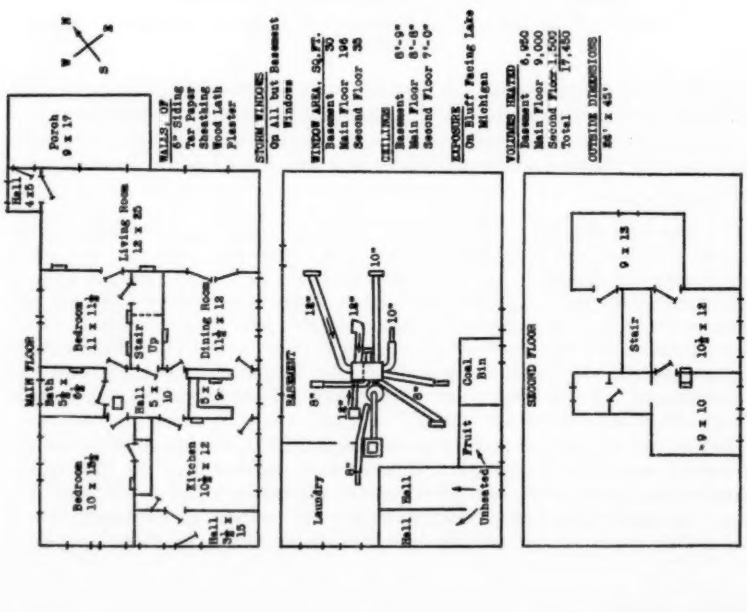


FIG. 5. RESIDENCE FLOOR PLANS

TABLE 1. SUMMARY OF MARCH AND APRIL OPERATING DATA

Averages of Daily Readings

Month	March	April
Coke burned, lb	111	73
Ashes removed, lb	8.6	4.8
Humidity water evaporated, lb	39	28.8
Energy used by fan, kwhr	0.66	0.3
Number of thermostat operations	26.5	20
Flue gas analysis		
CO_2	9.0	10
O_2		10.15
CO		0.20
Relative humidity in rooms, per cent	43	49
Outdoor temperature, degrees Fahr	32.7	43.8

TABLE 2. COKE AND ASH ACCOUNTING

Coke Delivery No.	1	2	3
Date first used	Feb. 21	Mar. 16	Apr. 8
Lb coke, as fired	2,417	2,345	1,925
Moisture, per cent			
As received		21.5	10.0
Last fired		13.0	5.0
Average	17[a]	17.0	7.5
Lb coke, dry	2,000	1,950	1,780
Ash			
By analysis, dry, per cent	9.5	9.5[a]	6.7
Refuse			
Lb, dry	202	155	121.5
Per cent, combustible, dry	8.2	6.2	7.4
Ash only, lb	185	145	113
Ash only, per cent of coke	9.25	7.4	6.3
Difference, analysis versus ash weight, per cent of coal weight	+0.25	+2.1	+0.4

[a] Assumed.

TABLE 3. COKE ANALYSES

Coke Delivery No.	1	3
Producer	Gas Company	Coke Company
Volatile matter, per cent	4.88	1.27
Fixed carbon, per cent	85.62	92.03
Ash, per cent	9.5	6.7
Sulphur, per cent	0.81	.....
Btu per lb, as received	10,400	11,700
Btu per lb, dry	13,255	13,000
Btu per ton delivered, million	24.7	22.5
Cost per ton	$8.50	$9.50
Cost per million Btu, cents	34.4	42.2

reducing CO losses. Continuous fuel feed onto a thin fuel bed is the best solution.

The continuously high combustion rate during the 10-hour capacity test caused the formation of 2.6 per cent CO after 1½ hours of test. It was necessary to admit air above the fire to eliminate this loss. Since continuously

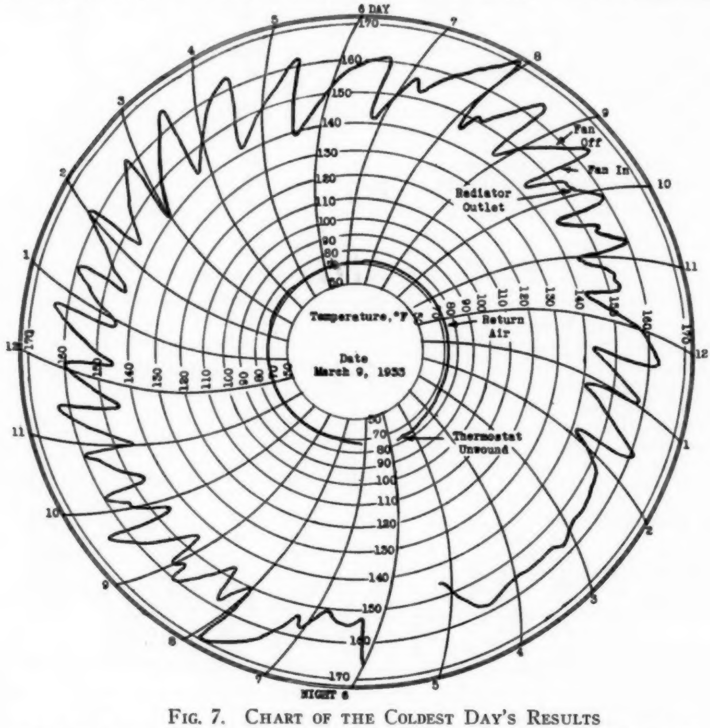


FIG. 7. CHART OF THE COLDEST DAY'S RESULTS

+ 5 F and a 30 mile per hour northwest wind most nearly taxed the capacity of the apparatus. Note uniform return air temperature, which is a good index of average room temperature. The thermostat operated 49 times on this day.

high ratings are so seldom carried for periods longer than an hour, the annual loss from this cause is slight. It does indicate one limitation of combustion rate in the installation, however.

Re-charging an empty hopper on another occasion caused two *puffs* which lifted the hopper cover slightly for pressure relief. Unless over-fire air was admitted after hand charging the previous warm-air furnace, this trouble would occur quite consistently.

Hopper Performance

Burning-back of the fuel into the hopper positively never occurred. In fact, inspections generally showed no incandescent fuel near the boiler opening, and

in most cases only the more distant half of the fuel bed was afire. Air-tightness of the hopper was undoubtedly responsible. Leakage into the hopper through its cover was observed to cause an appreciable increase in excess air and tightness caused as high as 14 per cent CO_2 flue gas to be present above the fuel in the hopper. Drying of the fuel caused condensation of water vapor on the walls of the hopper, which finally caused undetected caking and temporary failure of the gravity feed principle. Removal of the caked fuel, and monthly cleaning, prevented further trouble.

Boiler Water Control

On several occasions of rapidly increasing ratings, steam was formed in the boiler and released to the room through the vent provided. Evidence of expansion-tank water spillage onto the basement floor told what had happened. The water-temperature regulator has a thermal lag, but succeeds in adequately checking the fire without permitting steam-binding of the radiator or any interference with automatic operation. The float-operated make-up valve automatically replenished the lost water.

Several dissolved-oxygen measurements of the boiler water showed that it contained only one per cent as much oxygen as tap water (0.05 cubic centimeter per liter versus 6.0 cubic centimeters per liter). At all times the boiler-water is crystal clear, which indicates an absence of corrosion. Under these conditions, the thin copper tubes of the automotive radiator are expected to last indefinitely, and never to become clogged with sediment.

Year-round Fire Maintenance

Service-water heating during summer months not only affords inexpensive hot water but it eliminates the need of starting several fires each year for mild weather comfort. On two occasions, one in June and one in September, cooling and heating were needed and actually used in reverse sequence to the seasonal trend. Storage of heat in the boiler water affords an instantaneous space-heating supply. Cooling and heating operations can be alternated in a minute at any time simply by operating three valves and starting or stopping the fan. Morning *chill* can be dispelled immediately, and without causing temperature over-runs.

Anthracite coal of buckwheat size was used satisfactorily for a week of service-water heating, but was discontinued because of its greater dustiness than coke. Pea-size anthracite burned satisfactorily for mild-weather heating. It is improbable that the combustibility of the larger sizes of anthracite is sufficiently high to maintain combustion over long periods at the very low rate required for service-water heating.

Smoke pipe CO_2 readings when using coke fuel for service-water heating averaged above 5 per cent, and the smoke pipe outlet appeared at room temperature to the touch. To furnish a continuous supply of service water for four individuals, at an average temperature of about 135 F, 450 lb of coke, costing $1.50, was burned per month. The year-round hot water cost will average less than $1.00 per month, for the boiler radiation losses are rightfully charged to space heating for 8 heating months.

Humidity Conditions

Though measurements showed about only 45 per cent average relative humidity a higher moisture content of room air would have been undesirable.

Moisture collecting on the inside of storm windows during relatively warm weather indicated that similar wetness inside building walls might be anticipated in colder weather. Storm windows, though equipped with gaskets to make them relatively air-tight, frosted very considerably during the coldest weather.

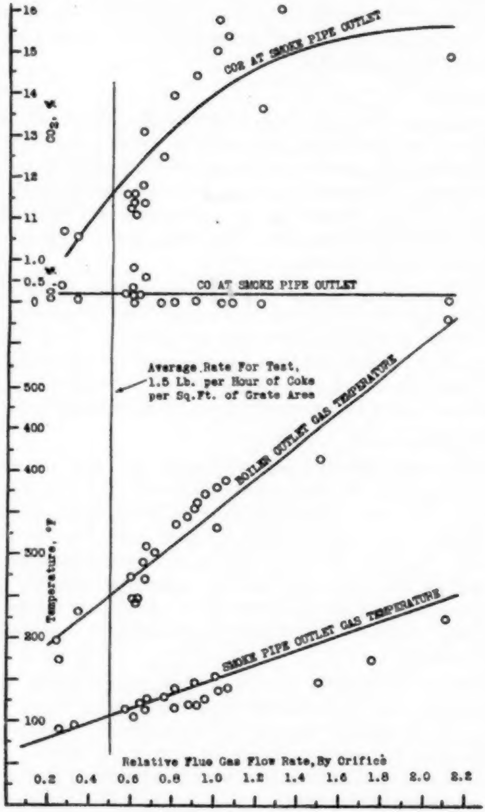


FIG. 8. SERVICE TEST OF SMALL HOT-WATER BOILER

Outdoor temperature 45 F. Boiler cleaned before test. Efficiency, crediting radiation losses, in excess of 90 per cent.

Previous experience with high humidity suggests that the present amount will cause condensation on some internal walls during zero weather. This indicates that the present equipment has adequate capacity. Tables 1 and 2 show the quantities of water evaporated per day. To accomplish this, 4.5 per cent of the fuel consumed on an average heating season day was required for evaporating the humidity water and 80 lb of fuel was burned on the average day, and

31 lb of water evaporated. Initial charging of the humidity pan with a small amount of chemicals prevented scale formation and corrosion.

DISCUSSION OF SPECIAL TEST RESULTS

Capacity

The unit operated 10 hours at an average heat output from the radiator slightly in excess of 100,000 Btu per hour. During this time it controlled itself automatically, and the boiler water temperature averaged 206 F. (See Table 4 and Fig. 9.)

The other two special tests were performed at approximately 25 per cent and 50 per cent of maximum capacity. During these tests the boiler was controlled by the room thermostat, which operated the drafts about hourly. As a result, boiler rating varied over wide limits throughout the tests, thus simulating actual operation. The fan was kept running constantly for the purpose of measuring air flow and heat output more accurately.

Radiator Performance

A unit-heat-transfer rate of 8.5 Btu per hour per sq ft per deg F, nearly 6 times the 1.5 rate of cast-iron hot-water radiators, was obtained from the radiator during the capacity test. Counterflow design, and adequately high air velocity, made this high value possible. A 167-sq ft radiator costing only $26 was made to do the work of 650 sq ft of standard radiators costing $150 without piping.

McAdam's *Heat Transfer*, p. 226, affords a means of predicting automative-radiator performance which agrees fairly well with the results obtained.

Efficiency

These special tests support the continuous operating data in indicating an average operating efficiency during the heating season of at least 90 per cent. CO_2 averages 10.5 per cent (see Fig. 6), and the accompanying flue-gas temperature at the smoke-pipe outlet is 160 F. Combustible in the ash averages 6.5 per cent, or about ½ per cent of the fuel. Reference to Fig. 9 shows efficiencies calculated from heat balance data over the range of loads; figured on the basis of total heat output to the radiator, humidity pan, and water heater, and on the second basis of total heat output including boiler radiation and smoke-pipe heat supply to the basement. The latter efficiencies are reported because the basement requires heating, since the air-flow casing and leader pipes have but slight heat loss. The former efficiencies were figured to permit comparison with measured efficiencies.

Measured efficiencies were higher in every case than heat-balance calculations indicated might be attained because of uncontrollable differences in unconsumed fuel in the boiler at the starting and stopping of tests. Because of the nature of the installation, starting from cold and quenching at the end of each test was not practical and would not represent true operating conditions. Power plant experience has indicated the heat balance method entirely reliable, especially when low losses occur, as in this case. The measured efficiencies are not reliable even though a special effort was made to obtain accurate input and output measurements during the maximum capacity test. The measured efficiency is about 10 per cent too high.

It is interesting to note that only a 9 F temperature drop was averaged between the radiator outlet and the registers during the capacity test. This shows the value of forced circulation for high outputs and proves the need of boiler and smoke-pipe heat losses to heat the basement adequately. Study of

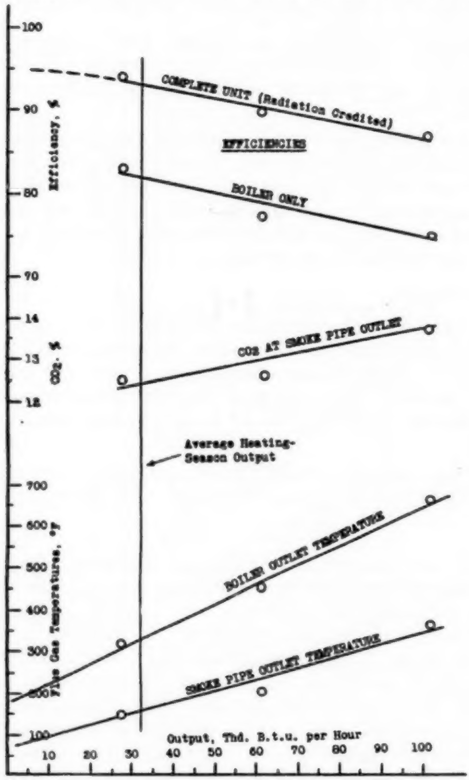


FIG. 9. RESULTS OF SPECIAL TESTS

Tests made with off-and-on thermostatic control during regular, typical residence service, are closely indicative of annual results. Because of low heat losses from the air conditioner, basement heating is required, and the upper efficiency line applies.

the heat balance data will show an analysis of the various losses and their magnitude. Coke is conducive of high boiler efficiency because of its low hydrogen loss.

COOLING EXPERIENCES

Occurrence of 101 F weather in June emphasized the comforts of residence cooling. Absence of awnings and tree shade, and continuance of the warm

period for several days, afforded an adequate test of the cooling cycle. Early-morning room temperatures of 78 F rose to 81 F by evening when water-flow was not allowed to exceed 1,000 lb per hour (0.9¢ per hour). Later experience proved that greater water flow improved the radiator performance more than expected, and 79 F room temperature was later maintained positively constant on a bright 95 F (maximum) day with 60 F inlet water. The outside temperatures cited are those of the local weather bureau. Occupancy consisted of two adults and two children. A cooling capacity of 12,000 Btu per hour was proved to be as adequate as a 75,000 Btu per hour heating capacity. This experience indicates that cooling calculations for the residence concerned can be made with the same heat-per-degree-Fahrenheit as employed for heating, and that 10 F average cooling is an adequate allowance. Table 5 shows typical cooling data for two days in June.

An effective temperature of 75.5 was maintained as a daily average throughout the summer with tap water from 50 F to 60 F. The A. S. H. V. E. Guide indicates 75 deg as the upper boundary for the Summer Comfort Zone.

Thermal storage of the residence structure acts as a relatively large flywheel to assist the performance of a cooling unit. The previously mentioned 3 F rise during the hottest day was entirely acceptable, and was regained before the next morning by continued operation of the unit throughout the night at reduced rating, though the night was relatively warm. Approximate calculations of the heat-storage capacity involved accounted for the result, and indicates the practicability of almost entirely neglecting sun effect in installations where cooling equipment is rather expensive. In this particular case, the radiator could be made twice as large at relatively little expense, and high water flow could be employed to obtain ideal results, but the actual results obtained were satisfactory.

Dehumidification did not afford the 50 per cent relative humidity median suggested by Comfort Chart in The Guide for 70 per cent was averaged. On one 36-hour run, 2 lb per hour of condensation was collected, and inside air averaged three-fourths the water-vapor content of outside air. Water velocity through the radiator affected condensation rate appreciably. Drainage of condensation from the closely-spaced fins and tubes was satisfactory, though about one-sixth of the openings were stopped by droplets. Best radiator performance was 10 per cent below its unit heat transfer of 8.5 Btu per hour per square foot per degree Fahrenheit when heating. Approximately 5 per cent reduction was expected because of the lower fluid temperatures. Condensation heat transfer would have been a net gain had not some of the air passages clogged.

Room air seemed much like the enjoyable fresh air from Lake Michigan, or like that during a cool evening or after a rain, all of which are high in humidity. With adequate cooling, filtration and circulation, 70 per cent humidity in air may be more acceptable than 50 per cent humid air. At least, it is more typical of average summer air.

Thermostatic control of the cooling cycle did not prove desirable because of the need of previously closing all windows and doors when morning temperatures indicated that a warm day was in prospect. Automatic control would create much extra operating cost unless assisted intelligently. There is no need of modulating control during the day, for room temperatures change very

TABLE 4. SPECIAL TESTS

Capacity, Approximate, Per Cent	25	50	100
General			
Date, 1933	March 26	March 19	April 9
Duration, hours	8	10	10
Outdoor temperature	45	26	40
Temperatures			
Air heated			
Radiator outlet	98	136	168
Radiator inlet	69	72	72
Radiator rise	29	64	96
Flue gases			
Boiler outlet	320	451	667
Smoke-pipe outlet	128	202	361
Boiler water			
Radiator inlet, average	142	172	206
Radiator outlet, average	....	....	177
Service water heater			
Inlet	....	....	206
Outlet	....	....	189
Service water			
Outlet	....	....	125
Inlet	....	....	108
Basement temperature	66	64	....
Flue Gas Analyses			
CO_2	12.5	12.6	13.7
O_2	8.25	....	6.6
CO	0.15	....	0.3
Total Quantities, Lb			
Humidity water evaporated	....	....	26
Fuel used	21.5	49	109
Refuse			
Per cent combustible	7.33	6.2	5.25
Air Flow			
Cfm above, radiator	935	935	1175
Cfm at registers	....	....	1291
Cfm average	....	....	1233
Fuel			
Total fired, lb	21.5	49	109
Lb per hour	2.69	4.9	10.9
Analyses			
Fixed carbon, per cent	....	....	92.03
Ash, per cent	....	....	6.7
Moisture, per cent	15	18.0	8.0
Btu per lb, as fired	10,600	11,000	11,900
Btu per hour fired	28,500	54,000	128,500
Heat Output, Btu per Hour			
By radiator	27,800	61,500	102,000
At registers	....	....	101,000
To heat service water	1,000	1,500	2,000
To evaporate humidity water	1,000	1,500	2,700
Total, at radiator	29,800	64,500	106,700
Total, at registers	....	....	105,700

TABLE 4. SPECIAL TESTS—*Continued*

Capacity, Approximate, Per Cent	25	50	100
Heat Balance and Efficiency Data			
Boiler only			
Losses, due to			
Moisture in fuel	1.1	2.1	0.93
Hydrogen (1 per cent)	1.0	1.04	0.99
Dry flue gas	10.2	15.50	18.5
Moisture in air	0.15	0.25	0.35
CO	1.0	1.5	2.1
Combustible in refuse	0.5	0.5	0.41
Radiation	3.1	1.9	1.5
Total	17.1	22.8	24.8
Calculated efficiency, per cent	8.29	77.2	75.2
Boiler and Smoke Pipe			
Losses, due to			
Moisture in fuel	1.0	1.9	0.80
Hydrogen (1 per cent)	0.9	0.95	0.85
Dry flue gas	2.4	5.4	9.1
Moisture in air	0.05	0.12	0.17
CO	1.0	1.5	2.1
Combustible in refuse	0.5	0.5	0.41
Radiation (Useful)	0	0	0
Total	6.0	10.4	13.4
Calculated efficiency, per cent	94.0	89.6	86.6

slowly. Again, structural thermal capacity is important. Cooling and heating duty can be likened to relative motive requirements for marine and automotive duty, respectively. Residence cooling equipment can be *baseload,* this experience indicates.

All registers were used for cooling without change. Return air averaged 2 F lower than air at the breathing level, and 3½ F difference existed from floor to ceiling. Register velocity of 400 fpm caused the cooled air to ascend to the breathing level at about 45 deg angle, after which it descended. Its velocity could readily be noticed at lower levels across the 12-ft to 14-ft rooms. This experience of cooled air falling was found to be of great assistance in cooling an auditorium during the past summer. Discarding of cooling water onto the residence lawn proved a practical and desirable procedure. Irrigation requirements were not exceeded during the 312 hours of cooling found necessary. Table 6 summarizes cooling data for the summer.

Considering the extremely acceptable relief the cooling equipment afforded at the modest operating cost of 1.27¢ per hour and nearly negligible investment cost, the system can certainly be considered a practical success under the particular local conditions prevailing in Milwaukee.

Records of tap-water temperatures at an outlying point in Madison, Wis., prove that results would be equally satisfactory in that city, which distributes 53 F well water at 1.2¢ per 1,000 lb. When studying tap water temperatures for cooling, it is extremely important that they be taken during periods of continued warm weather, when residential water use is high and distribution

TABLE 5. COOLING TESTS

(June 7 and 8, 1933)

Time	Air Temperatures, Degrees Fahr: Outside SW	Outside NE	Outside SE	Rooms: Living	Rooms: Hall	Radiator Out.	Basement	Attic	Water Temp. Out	Water Temp. In	Lb Condensate	Meter Readings: Water Cu Ft	Meter Readings: Electrical Kwh
6/7/33													
A.M. 7:00	79	..	79	77	76.5	66	72	..	66	54	..	....	....
7:30	80	..	..	77.5	76	66	71.5	81	66	55	..	846.5	37.7
8:00	81	..	..	77.5	76	65	71.5	..	..	..	..	853	....
10:00	..	..	..	77.5	76.5	65	71	..	66	55	..	884	....
P.M. 12:30	88	..	..	78	76.5	65	70.5	..	66	53	..	910	....
1:00	93	..	..	78	76.5	65	70.5	95	63	53	..	....	....
5:30	100	91	..	81	79	64	71	102	62	52	..	966	....
7:45	80	81	..	81	80	62	71	92	64	51	..	1012	....
9:30	..	..	..	...	...	...	...	..	...	51	..	1029	38.6
6/8/33													
A.M. 7:00	77	..	79	78.5	76	66	70.5	..	65	54	..	1120	39.4
P.M. 1:00	90	..	92	80.5	78	63.5	71	100	63	52	..	1125	39.8
6:00	..	..	86	82	81	63	72	96	63	52	..	1318	....
7:30	..	..	..	...	...	...	...	..	...	..	..	1328	40.2
Total 36.5 hours	..	..	..	...	...	...	...	..	..	..	75	525.7	2.5
Average 1 hour...	85	86	84	79	77.5	64.5	71	95	64.4	53	2	900	70

UNITS	LB	WATTS
Cost per hour, ¢	0.81	0.21
Total, ¢	31	7.5
per day, ¢	15.5	3.8
per day, water and electricity, ¢	19.3	...

Summary of Cooling Tests

Humidity Results

Dehumidification, lb per hour, average 2

6/7/33, 5:30 P.M.

	Inside	Outside
Temperatures, degrees Fahr		
Wet bulb	72.5	81
Dry bulb	79.5	91
Per cent relative humidity	73	66
Moisture content, grains per lb air	112	148
Relative moisture content, per cent	75	100

Cooling Results

Duration of continuous test, hour	36.5
Average outside air temperature, degrees Fahr	85
Average inside air temperature, degrees Fahr	78
Average temperature difference, degrees Fahr	7
Maximum outside temp., deg Fahr (Weather Bureau)	94[b]
Maximum temperature difference, deg Fahr	16
Water flow, lb per hour	900
Electricity used, watts	70
Average cost per hour, total	1¢

[b] 82 per cent sunshine on June 7; 99 per cent on June 8.

TABLE 6. SUMMER COOLING DATA AND COSTS

Month	June	July	August	Sept.	All
Hours of cooling	186	88	16	22	312
Cooling water, cu. ft.	2,793	1,671	340	495	5,299
Cooling water, lb. per hour	940	1,190	1,330	1,390	1,060
Cooling cost, $	1.68	1.00	0.20	0.30	3.18[c]
Cooling cost, ¢ per hour	1	1.1	1.25	1.35	1.02
Electricity, 26 kwhr at 3¢	....	....	....	....	$0.78
Total cooling cost for summer	....	....	....	....	$3.96
Total cost per hour	....	....	....	....	$0.0127
Estimated total cost for normal summer	....	....	....	....	$6.00

[c] $5.84 was total billed for period of May 12 to September 26, 8,900 cu ft at 6¢ per 100 cu ft plus 50¢ service charge. 4,900 cu ft was used the previous summer.

TABLE 7. CAPACITY AND DESIGN DATA OF EQUIPMENT

BOILER: Cast iron, round sectional, 280 sq ft 8-hour rating, 825 sq ft rated capacity, 18 in. diameter. 1.76 sq ft grade. 48 in. high, 22 in. outside diameter. Approximately 18.75 sq ft heating surface. Water capacity 90 lb. Total weight, 700 lb.

Covering: ¾ in. plastic cement having 0.8 Btu per hour per sq ft per degree Fahrenheit per 1 in. conductivity, painted aluminum bronze.

Grates: ⅛ in. opening, 3 in number.

SMOKE PIPE: 9 in. diameter, 10 ft long, air-tight 24-gage sheet metal, painted black.

RADIATOR: 20½ in. × 23⅜ in., automotive-type, fin and tube, 167 sq ft.

Fins: 6 per inch, 3⅝ in. × 23½ in., 0.005 in. thick, 140 total.

Tubes: ⅛ in. × ¾ in., 0.008 in. thick. 7/16 in. spacing, 4 rows deep, 2 per pass, 184 total tubes.

Water boxes: 18 gage brass. Inlet water box has internal baffle. 2-in. standard pipe connections.

2.25 sq ft air flow area, 5.9 sq in. water flow area per pass. 26 per cent direct surface, 74 per cent finned surface.

MOTOR: 1/20-horsepower, 1140 rpm, 110 volts, unit heater type. Totally enclosed, wick oiling, 57 per cent efficiency, 55 per cent power factor, 1.07 amps, 65 watts at full load. Rubber mounting.

FAN: 4 blade 16-in. propeller type.

FILTER: 4-20 in. x 20 in. x 2 in. oiled glass-wool units in parallel.

HUMIDITY PAN: 6 in. x 20 in. x 20 in., 12-gage sheet pan. 2—16-in. lengths of 1-in. pipe and return bend for heating coil.

LEADER PIPES: 2.5 sq ft flow area, composed of 34 ft of 8-in. pipe and 15 ft of 10-in. pipe.

COLD AIR RETURN PIPES: 2.35 sq ft air flow area, composed of 21 ft of 12-in. pipe.

FUEL HOPPER: 4.6 cu ft, 125 lb normal charge.

AIR FLOW CASING: 2 ft x 4 ft x 5 ft high outside. Wall-board on 2-in. x 2-in. pine frame. Painted aluminum bronze. Lined with ⅜ in. of felt.

THERMOSTAT: Spring motor. 50 operations per winding. Operates on bell transformer.

temperature rise is smallest. September cooling in Milwaukee was as satisfactory as in June, though the pumping station is 6 miles distant.

It is emphasized that these cooling results were obtained with an efficient, counterflow radiator and reasonably cool tap water. Careful attention to

radiator design does much to make tap-water cooling acceptable. Air can be cooled within 5 F of the entering water temperature, and the outlet water temperature can be 5 F higher than the cooled air by proper design.

For the average summer in Milwaukee $6 summer cooling costs are estimated. Though June was 10 F warmer than ordinarily, July and August weather was favored with an unusual preponderance of winds from Lake Michigan which limited daytime temperatures, causing only $4 cooling cost. (See Table 6.) Milwaukee's incremental water and electricity costs of 6¢ per 100 cu ft and 3¢ per kwhr are low, but doubling or trebling of the actual cooling costs would not make cooling impractical.

TOTAL ANNUAL COSTS

Using data from the previously installed warm-air furnace and from the present installation, costs compare as follows:

Installation	Warm-Air Furnace	Air-Conditioning Unit
Initial cost, installed	$200.00	$250.00
Annual fixed charges, 12 per cent	24.00	30.00
Operating Costs		
Heating		
Tons coal	10.0	8.5
Cost per ton, $	8.50	8.50
Coal cost, $	85.00	72.25
Electricity, 133 kwhr at 3¢	......	4.00
Total cost, $	85.00	76.25
Cooling		
Water cost, $	......	4.85
Electricity cost, $	......	1.17
Total cost, $	......	6.02
Service water heating during summer (4 months)		
Gas cost, $0.75 per M, $	6.00	......
Coke, cost, $	......	4.00
Maintenance		
Annual average, $	15.00	5.00
Total annual operating and fixed charges	$130.00	$121.27

Better thermal economy and lesser maintenance of the newer unit more than pay its summer operating costs. The advantages of having year-round conditioned air, with less attendance, are incidental.

CONCLUSIONS

Design and actual operating data are included herein to show that satisfactory year-round air conditioning can be practiced under the particular existing conditions in Milwaukee at no greater total operating and investment cost than simple warm-air heating. High boiler efficiency, low cooling cost, and moderate equipment costs make this result possible.

No. 978

COMFORT COOLING WITH ATTIC VENTILATING FANS

By G. B. Helmrich * and G. H. Tuttle † (*NON-MEMBERS*)
Detroit, Mich.

EXPERIMENTS in residential cooling, initiated in the summer of 1932, were continued during this past summer, and operating data were gathered on three residential cooling and ventilating installations which were directly sponsored by the Detroit Edison Co. This paper summarizes the season's operating experience with these installations and describes certain features of design and construction which, it is believed, will be of special interest at this time.

Cooling by Attic Ventilating Fans

The costs of cooling installations which have sufficient capacity to cool artificially a moderate-sized residence are still high enough to restrict their use to a comparatively few people, while the desire for more comfortable conditions in the home during the hot summer season is almost universal. If a simple and relatively inexpensive means can be made commercially available, whereby at least the sleeping rooms may be made more comfortable at night, there unquestionably would be a broad market for such equipment.

This suggests the development of a method by which the outdoor air can be drawn through the rooms at such times as the temperatures are lower than those indoors. A study of the summer temperature records for Detroit shows that, almost invariably, the outdoor temperature drops below the inside temperature at around 7 to 8 p. m., and stays below for approximately 12 hours. The attic ventilating fan seemed to give such promise as a means of accomplishing this purpose that three types of residential installations were made during the past year and a half and considerable experience has been gained with their operation. The attic ventilating fan is by no means a new idea, but its possibilities seem to have been quite overlooked by nearly everyone.

In addition to providing more comfortable conditions for the second floor of the home, it was felt that an attic fan could be used to pull air through the first floor rooms during the early evening and thereby accomplish an appreciable improvement to the inside air condition usually existing after a hot summer day. It is then just one step further to install an attic fan of sufficient capacity

* Engineer, Construction Bureau, The Detroit Edison Co.
† Heating Engineer, The Detroit Edison Co.
Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by G. B. Helmrich.

to pull a large volume of air through both the first and second floor rooms and so cool down the building structure during the night that, with no artificial cooling during the following day, the conditions will be tolerable. It was with the thought that the attic ventilating fan might serve as a substitute for artificial cooling in many residences, where a moderate improvement to comfort would be most welcome, that this experimental work was undertaken.

THE FAN INSTALLATIONS

The installations made in the Detroit residences were of three distinct types. One was a standard commercial propeller fan, the second a centrifugal multi-

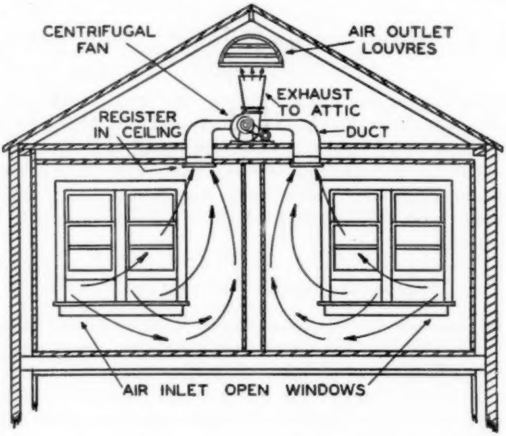


FIG. 1. CENTRIFUGAL MULTI-BLADE FAN INSTALLED WITH EXHAUST DUCTS

blade fan, and the third a centrifugal, steel plate, curved blade fan, installed without a scroll. The essential requirements for cooling systems of this kind are quietness of operation and low first cost. Fan efficiency is unimportant because of the comparatively short season of use.

Centrifugal Fan

A multi-blade centrifugal fan was installed in the attic space of an eight room house of brick and stucco construction. The suction side of the fan was connected to exhaust ducts leading to grilles placed in the ceilings of two bedrooms, the air being drawn through the open bedroom windows and discharged into the attic space where it flowed to the outdoors through louvres placed in the gables. The general arrangement is shown in Fig. 1 and the principal data for the installation are shown in Table 1.

Propeller Fan

A 10-blade propeller fan of the disc type was installed at the head of the attic stairway of a six room house of ordinary frame construction. The fan

was mounted over the attic stair well in such a manner that the entire attic stairway is placed under suction. The air is drawn through the open windows of the three bedrooms into a central hallway, and thence up the attic stairs where it is discharged into the attic space and allowed to find its way to the

TABLE 1. COMPARISON OF ATTIC VENTILATING FANS IN RESIDENCES

Type of Fan	Centrifugal Multi-Blade (Exhaust Duct System)	Propeller	Centrifugal Steel Plate Curved Blade —No Scroll
Diameter of Wheel—Inches	18	21¼	36
Capacity — cfm (as measured)	2754	3060	5040
Fan Speed—rpm	300	900 600	320
Blade Tip Speed—fpm	1415	5000 3300	3010
Motor Drive	½ hp. Capacitor Start and Run—1140 rpm—V-Belt Drive	Direct Connected to a ⅙ hp. 900 rpm A-C Motor	⅓ hp. Capacitor Start and Induction Run—1715 rpm—V-Belt Drive
Static Pressure of Fan —Inches of Water	0.10[a]	0.038[b]	0.11[a]
Volume of Space Ventilated—cu. ft	4600	5000	5800
Rate of Air Renewal—Changes per hour	36	37	52
Fan and Motor Mounting	Sponge Rubber insulated Wood Platform	Bolted to Wood 2x4's faced with strips of felt	Welded Angle Iron Frame supported at four corners on 2 in. x 2 in. x 1 in. sponge rubber pads
Power Input—Watts	320	252	375
Cfm per Watt Input	8.6	12.1	13.4
Power Consumption for Season 1933—kwhr.	48	14	....
Hours of Operation—1933	150	55	....

[a] Static pressure of fan is computed to be the static pressure at fan outlet plus static pressure (draft) at fan inlet minus velocity pressure at fan inlet; draft gage mounted in the attic in all cases.
[b] Static pressure readings, on the suction and discharge side of the fan, were taken where air velocities were negligible, consequently no correction for velocity head is made; draft gage mounted in the attic.

outdoors through open windows placed at either end of the attic. The general arrangement is shown in Fig. 2 and the installation data are given in Table 1.

Centrifugal Steel Plate Fan

In an endeavor to avoid some of the noise experienced with propeller fans, and at the same time overcome the obstacle of higher installation costs which are inherent in a system requiring duct work connected to a centrifugal fan, a rather unique fan assembly was devised. This assembly, shown in Fig. 3, consists of a single inlet, backward curved blade, steel plate type, centrifugal

fan wheel, mounted on a light angle iron frame which supports the fan wheel, shaft, two bearings, V belt drive, and motor. No fan scroll was used and the inlet was placed close to a circular opening cut in a plaster board partition which served to divide a small attic space into a suction and discharge chamber. As in the case of the propeller fan installation, air is drawn up the attic stairway, into the fan wheel, and discharged freely from the blades into a small attic space, from whence it flows through open attic windows to the outdoors. The installation data for this fan are also to be found in Table 1.

OPERATING RESULTS

The operating experience gained with these three installations during this past summer has shed considerable light on the possibility of popularizing this

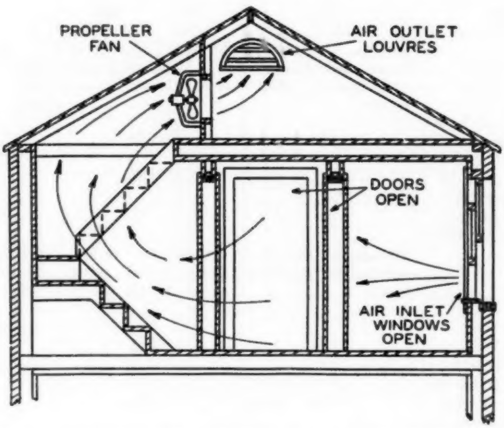


FIG. 2. PROPELLER FAN INSTALLED IN ATTIC STAIR WELL

type of comfort cooling, and the data and experience will be discussed in the order in which the installations have just been described.

Centrifugal Attic Ventilating Fan

The centrifugal fan illustrated in Fig. 1 was operated about 150 hours during the past summer, and the occupants of the house were well pleased with the results obtained. The temperatures in the two bed rooms to which the fan is connected were very responsive to the action of the fan, and the characteristic temperature curve for the larger of the two rooms is shown in Fig. 4. It will be seen that the inside temperature can be made to follow the declining outdoor temperature curve very closely, the temperature at midnight being only about 2 deg above that for the outdoors. This is in contrast to the conditions obtaining when the fan was not used, as the curves show the indoor temperature to be at least 8 deg higher than the outdoor at midnight, and

additional temperature records show this difference to have been as high as twelve degrees on some occasions.

Anemometer readings taken at the ceiling grille showed an air velocity of about 330 fpm, corresponding to 1294 cfm passing through this room. As the volume of the bed room was 2180 cu ft, this rate of air flow provided about 36 changes of air per hour. With a total air flow of 2754 cfm through the fan, the static pressure of the fan was 0.10 in. water. The efficiency of cooling [1] at this rate of air change is shown quite effectively in Fig. 5. The indoor and outdoor temperature drop has been plotted at one hour intervals between the point at which the two temperature curves cross and the point at which the drop is a maximum; the latter being the point at which the actual tem-

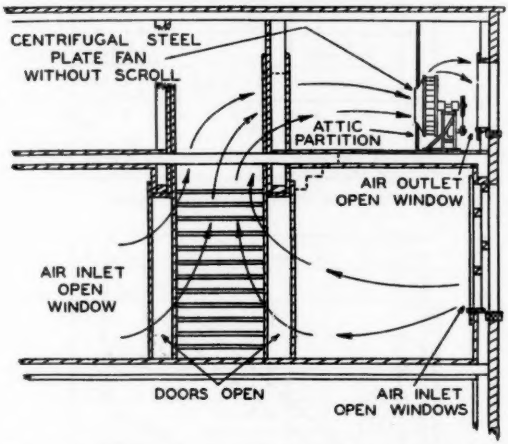


FIG. 3. CENTRIFUGAL CURVED BLADE STEEL PLATE FAN INSTALLED WITHOUT SCROLL

peratures reach their minimum valve. These data were taken from Fig. 4 for the evening of June 27. It is evident that sufficient outdoor air was being drawn through the bed room to make the indoor temperature drop quite uniformly with the outdoor, with a differential between the two temperatures of about 2 deg. On the previous night, the night of June 26, the fan was not operated, and natural ventilation was depended upon to reduce the inside temperature. The curve for natural ventilation, shown in Fig. 5, indicates a very slow drop in indoor temperature in the early evening hours, when interpreted in terms of the actual temperature curves for June 26 as they appear in Fig. 4. Since the actual maximum drop in indoor temperature was less than half as great as that for the outdoor, and also less than half as much as the drop in inside temperature on the succeeding night when the fan was used, it seems fair to conclude that, in this installation at least, natural ventilation

[1] Calculated as suggested by Prof. A. P. Kratz and S. Konzo, Engineering Experiment Station, University of Illinois, Urbana, Ill.

is only about half as effective in reducing inside temperatures as is the case with fan ventilation. This contrast is even more striking if consideration is given to the fact that, on the night the fan was used, both the day and night outdoor temperatures were considerably higher than obtained on the previous

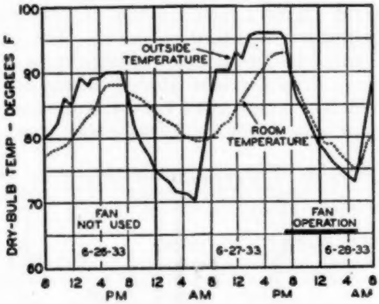


Fig. 4. Effect of Centrifugal Attic Fan Ventilation upon the Temperature of an Upstairs Room

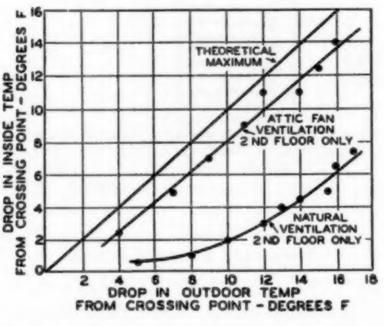


Fig. 5. Relation between Inside and Outside Temperature Drop Following the Point at Which the Temperatures Coincide

night when natural ventilation was employed. It should be pointed out, however, that in this installation, there was no opening into the attic, except through the ceiling grille, exhaust duct, and fan, so that natural ventilation was not aided by the stack effect of the structure, whatever real value that may have. The temperature records alone do not indicate the entire cooling effect produced. There was also a very noticeable air movement set up which decidedly augmented the temperature reduction.

Another illustration of the effectiveness of this fan is the comparison of bed room temperatures affected by the fan with the temperatures of another bed

room not connected to the fan. This comparison is shown in Fig. 6. The unventilated bed room has a northeast exposure while the ventilated bed room has a northwest exposure. It will be seen that the temperature curve for the

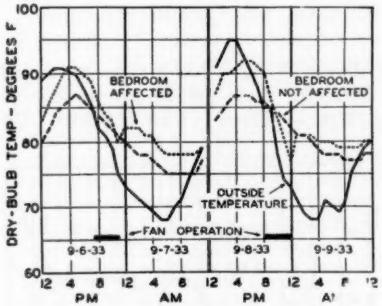


Fig. 6. Comparative Temperatures in Two Similar Bed Rooms, One Affected and the Other Not Affected by Attic Ventilation

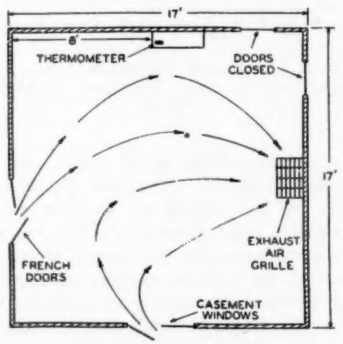


Fig. 7. Distribution of Air in a Bed Room Ventilated by an Attic Fan

ventilated bed room drops below the curve for the unventilated room, even though in both cases it starts from a much higher maximum.

Although the occupants of the room reported that the diffusion of the air drawn through it was quite satisfactory, it was decided to check this diffusion by moving the thermometer to various parts of the room on succeeding nights. The thermometer was located in the position shown in Fig. 7 when the temperatures were recorded which are shown in Fig. 6. When the temperatures shown in Fig. 4 were recorded, the thermometer was located near the

center of the room. Other locations were tried from time to time, and, except for the location near the ceiling grille, the recorded inside temperature drop showed very nearly the same relationship to the outdoor temperature drop in each case. This indicates a much better diffusion than was at first thought possible.

Propeller Fan

The propeller fan shown in Fig. 2 has a capacity, as determined by an anemometer, of about 3000 cfm when pulling air through the three bed rooms. This is equivalent to about 37 changes of air per hour. Under these conditions

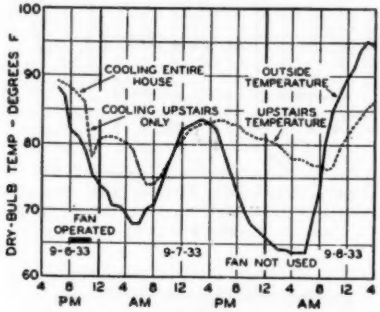


FIG. 8. EFFECT OF PROPELLER ATTIC FAN VENTILATION UPON UPSTAIRS TEMPERATURES

the static pressure of the fan is 0.038 in. of water. As the fan was operated over shorter periods than was the case with the centrifugal fan, the bed room temperatures did not fall nearly as far, but during the rather short periods of operation, usually from about 8 to 11 p. m., the cooling effect was just as pronounced, as is evidenced by the curves shown in Fig. 8. It will be noted that, during the fore part of the evening of September 6, the fan was used to pull air through both the downstairs and upstairs rooms. When operating in this manner air was pulled through the downstairs rooms and up the stairway at the rate of about 10 air changes per hour, while at the same time air was drawn through the bed rooms at the rate of about 20 changes of air per hour. The break in the upstairs indoor temperature curve indicates the point at which the downstairs windows and doors were closed, and from that time on the air was drawn through the bed rooms only. The rather sharp rise in indoor temperature, which takes place after the fan is shut down, is characteristic of these installations, and is undoubtedly due to the heating effect of the walls and ceiling. The inside temperature drop obtained when natural ventilation was used seems to bear about the same relationship to the temperature drop produced by the propeller fan as that obtained with the centrifugal fan previously described. When the fan was not used, the attic door was kept open as before, consequently the stack effect of the structure could be utilized. Although there was a noticeable natural draft up the attic

stairway, the cooling effect produced was in no way comparable to the results obtained when the fan was operated.

On several occasions the fan was used during the early evening to ventilate the downstairs, and quite satisfactory results were obtained. To obtain best results, of course, the bed room doors had to be kept closed. The fan was especially effective in removing tobacco smoke from the living room when a group of people were present. If the bed room doors were closed, tests showed that it was possible to pull sufficient air through the downstairs rooms to produce about 30 changes of air per hour. When the downstairs was closed and only the bed room windows were open, there was not enough air movement up the stairway to the second floor to register on the anemometer. This does not prove, of course, that there was no leakage of air into the lower part of the structure, as the anemometer readings had to be taken at a point where the area was rather large. None of the tests, however, indicated that the leakage of air into the structure was serious enough to have an appreciable effect on the cooling and ventilating capacity of the fan.

Centrifugal Steel Plate Fan

The fan assembly shown in Fig. 3 was not installed until the latter part of last summer, consequently reliable temperature data are not available. Capacity tests were made, however, and enough information was gathered to give assurance that the fan was capable of effectively ventilating the three upstairs bed rooms to which the center hall and attic stairs are connected.

On the first test the fan was operated at 390 rpm, and anemometer measurements showed an air movement up the attic stairway of about 5880 cfm. At this speed, however, there was considerable air noise, and for the second test the fan speed was reduced to 320 rpm. At the latter speed the fan handled 5040 cfm up the attic stairway at a static head of 0.11 in. of water. With the downstairs closed up, and the windows open in two bed rooms, the air flow through one of the bed rooms was 1580 cfm and through the other 2600 cfm. With the volume of the second floor equal to about 5800 cu ft this rate of air flow provided about 52 changes of air per hour. The measured air flow through the two bed rooms which were open was 4180 cfm, and the difference between this volume and the quantity of air actually handled by the fan can be accounted for by in-leakage of air through windows in other rooms and leakage up the first floor stairway. The air movement up the stairway, however, was not sufficient to register on the anemometer. This leakage and short circuiting of air amounted to about 860 cfm, or about 17 per cent of the total volume of air handled by the fan. With this rather high rate of air change it seems certain, considering the experience with the other installations, that this fan will accomplish very effective cooling.

Control of Fan Noise

Since the attic ventilating fan is usually located directly over the sleeping rooms, it is quite essential that every possible precaution be taken to reduce the fan and motor operating noises to a level which is not objectionable to the occupants of the home. In the case of the centrifugal fan installation, both fan and motor were mounted on a wood platform which was insulated from the

attic floor joists by strips of ½ in. sponge rubber, and rubber washers were used under all holding-down bolts. In spite of these precautions, the initial fan speed of 400 rpm proved to be too high for quiet operation, and it was reduced to about 300 rpm. It was also found desirable to limit the velocity of the air through the ceiling grilles to about 400 fpm in order to reduce the air noise. Motor noises have frequently proved troublesome, and the capacitor type of motor has been found particularly well adapted to this service because of its inherent quietness of operation.

The propeller fan installation described in this paper was not satisfactory from the standpoint of noise. At the normal operating speed of 900 rpm the blade tip speed was 5000 fpm and the air noise was quite disturbing. For that reason the fan usually was not operated after the occupants had retired for the night. Although strips of felt were placed between the fan casing and the wood studding to which it was bolted, this was not very effective in reducing the transmission of sound through the building structure. Experience thus far indicates that the top speed of a ventilating fan should not exceed 3200 fpm if quiet operation is desired.

CONCLUSIONS

1. It is quite practicable, by the use of either a centrifugal or propeller type attic ventilating fan, to very effectively cool the sleeping rooms in a moderate-sized house.

2. The downstairs portion of the house can be effectively ventilated if the fan is connected to an attic stairway in such a manner that the entire first floor can be placed under suction by way of the first floor stairs. It is not advisable, however, to leave the downstairs open after the occupants of the house have retired, as the flow of air through the bed rooms, and consequently the cooling effect in these rooms, is greatly reduced.

3. Wherever possible, the capacity of the fan should be such as to provide a minimum of about 30 air changes per hour. In very large homes, physical or commercial limitations may dictate the use of a proportionately smaller fan, providing about 20 changes of air per hour. This is still sufficiently high to accomplish reasonably satisfactory cooling.

4. Natural ventilation, induced by the stack effect of the structure resulting from the opening of attic doors and windows, is not nearly as effective in circulating air from out of doors as is an attic fan which provides about 30 changes of air per hour.

5. It seems reasonable to conclude, in the light of the past summer's experience, that attic fan ventilation for residences should prove to be a satisfactory substitute for artificial cooling during a large part of the summer season under climatic conditions similar to those in Detroit.

DISCUSSION

J. H. WALKER: In presenting this paper it was not intended to give the impression that the idea of the attic fan was new or original, for it has been advocated by a number of people for some years. The general conclusion drawn from our studies

is that it is a very inexpensive and a very practical way of obtaining a certain amount of cooling.

H. C. Bender: I have had considerable experience in this problem of noise control. A motor such as was used on the blower type units, mounted completely in rubber, has practically eliminated that source of trouble, but the matter of noise is one of very serious consideration. Women are very sensitive to this noise; therefore I think the manufacturers as a whole will have to give serious thought to this matter of elimination of noise in any equipment that goes into the home. It is hard to stop this noise even with furnace fans. We find that women will shut off the equipment rather than listen to the noise.

E. K. Campbell: In regard to the matter of noise from the propeller type of fan as compared with the centrifugal type of fan, any one who has had experience with the propeller type of fan knows that a 21-in. fan running fast enough to produce 5,000 cfm is going to be noisy wherever it is located, regardless of the manufacturer, or how many blades it has.

A 30-in. fan would reduce the speed. The noise would be eliminated, the horsepower would not be increased and the same volume of air would be obtained. That would handle the noise problem as far as the type of fan is concerned.

In regard to the diffusion of the cooling effect within the room, an exhaust fan pulling air from the ceiling of a room does not pull air directly to any extent from the floor. It pulls the heated air from the ceiling because that air lies up there due to its light weight. Consequently there is very little direct disturbing of the air near the floor due to an exhaust fan pulling air from the ceiling. Therefore the cooler air which was drawn in through the windows would settle to the floor and would spread across the floor very evenly diffused, and as it absorbed heat from the material of the room, it would rise to the ceiling and in turn be exhausted. You would find the same stratification of air due to temperature under those conditions that are obtained in a circulating or fan-heating system.

In a large installation we have found the variation in temperature on any given level in a room over 200 ft wide and nearly 300 ft long to be less than 3 deg, due to the fact that there was sufficient volume of air moving and stratification due to temperature was taking place constantly.

G. E. Olson: In Kansas City we had occasion to study a similar condition. We found that by locating the grilles on the outside walls of the room we had a better heat pick-up from the heat gain in the attic. In other words, we located the blower in the center of the attic space and had several grilles on the outside walls that let the air into the attic space and in that way picked up a lot of the heat that was normally storing up between the ceiling and the eaves, and the removal of that excess heat helped materially to drop that room temperature.

As the installation had a forced air heating system, we augmented the attic ventilation by turning on both the forced air heating system and the attic ventilating system at the same time. This produced a diffusion of the air through the rooms and, by having the air come across the second floor ceiling, we dropped the temperature in the attic, which was reflected immediately in the second floor rooms.

I should like to ask if there is any preference as to the location of the fans in the attic? Is it equally satisfactory to have them at the gable end of the house or in the center connected direct to the roof? We have found the center location preferable.

G. B. Helmrich: Due to our limited experience we cannot be certain which is the best location for the fan, but as far as our experience goes it indicates that we can get better results if we place the fan near the center of the attic. In this manner we can sometimes avoid the necessity of placing the entire attic under suction. We

made one experimental installation with a fan located at one end of the attic and our tests indicated that the leakage in that particular attic was rather high, being approximately 50 to 60 per cent. We are not certain about some of our test data, however, and we expect to go further with our experiments. I could describe this leakage in another way by saying that only about half as much air was being pulled through the bedrooms as was passing through the fan, according to our first tests, and this would indicate a 50 per cent leakage through the attic.

In the third installation described in the paper, the one using the centrifugal fan without the scroll, we determined the leakage by measuring the volume of air being pulled through the bedrooms with the lower part of the house closed, and comparing this quantity with the volume of air passing through the fan, as measured at the fan suction by means of an anemometer. This comparison indicated a leakage of only 17 per cent. We realize that we still have a great deal to learn about the leakage of air into structures when being ventilated by attic exhaust fans, but we feel that leakage can be measured and that it may amount to a very appreciable proportion in some installations.

No. 979

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE FOR THE SUMMER OF 1933

By A. P. Kratz * and S. Konzo † (*Members*), Urbana, Ill.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the National Warm Air Conditioning Association and conducted at the University of Illinois.

The results presented in this paper were obtained in connection with a continuation of the summer cooling investigation (1933) in the Research Residence[1] *(Fig. 1) at the University of Illinois, conducted by the Engineering Experiment Station under the direction of Acting Dean A. C. Willard, director of the Engineering Experiment Station and head of the Department of Mechanical Engineering. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Special acknowledgment is due the National Association of Ice Industries, the Utilities Research Commission of Chicago, Illinois, and the General Electric Company for active cooperation in this investigation. Acknowledgment is also due to M. K. Fahnestock, Special Research Assistant Professor, E. L. Broderick, Research Assistant, and A. F. Hubbard, Special Research Graduate Assistant, for active participation in the detailed work of the investigation.*

THE experience in the Research Residence, in which between 40 and 45 tons of ice were used during the summer of 1932,[2] has shown that complete artificial cooling of this class of structures by such cooling agents as cold water or ice, or by mechanical or chemical methods, may be a very expensive process unless some modification is made in the conventional or usual methods of operation or in the structure itself. The most obvious modifications in the structure consist of the use of insulation in the walls and ceilings, and the use of awnings at the sun exposed windows. The use of awnings was investigated during the summer of 1932, and the use of insulation was considered outside of the immediate scope of this investigation. Thus attention was directed toward some modification in the operating schedule.

* Research Professor, Engineering Station, University of Illinois.
† Special Research Associate, Engineering Experiment Station, University of Illinois.

[1] The Research Residence in Urbana, Illinois, was built, furnished and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934.

For the investigation during the summer of 1932 the windows and outside doors were kept closed as much as possible, and the ice plant was started whenever the effective temperature indoors reached a value of 75 F. During the periods of operation, both night and day, the effective temperature was maintained at approximately 72 F, corresponding to a dry-bulb temperature of from 78 to 80 F and relative humidity of about 45 per cent. However, a study of the temperature cycles indicated that the minimum outdoor temperature at night was always approximately 20 F lower than the maximum attained during the day, and that during a period extending from 9 p. m. to 6 a. m. the outdoor temperature was 7.5 to 10 F lower than that indoors. During

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILL.

this period the temperature of the inside surfaces of the walls was usually less than that of the inside air, so that all of the cooling of these surfaces had to take place by conduction through the wall to the outside air. Hence it seemed evident that if sufficient air at a temperature lower than that of the inside wall surface could be introduced, these surfaces could cool from the inside, and if the process was continued, the whole structure could be cooled, and the next day started with the entire structure filled with cool air, with the inside wall surfaces and contents at a temperature lower than would otherwise exist, and with the heat absorbing capacity of the material in the structure increased as a result of the lower wall surface temperatures. The investigation for the summer of 1933 was, therefore, undertaken with the object of determining to what extent the circulation of air taken from outdoors at night could be used to supplement ice cooling during the day, thus reducing the amount of ice required, and to what extent it could be used to eliminate the necessity for any additional cooling during the day. Incidental objects were to determine the relative merits of natural ventilation, of a fan in the forced-

air heating system, and of a fan in the attic as means for circulating the air taken in from the outdoors at night, and, if possible, to determine the most advantageous time for opening the windows and starting the fan.

DESCRIPTION OF THE RESEARCH RESIDENCE AND COOLING EQUIPMENT

The Research Residence, shown in Fig. 1, together with the forced-air heating system and the ice-cooling plant have been described in a previous paper.[2] For the purpose of this investigation the Residence was equipped with awnings at all east, south, and west windows and the sun parlor was

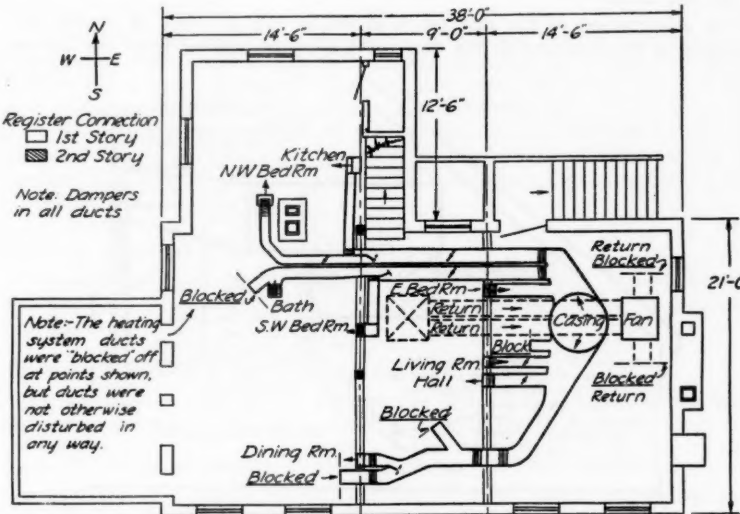


FIG. 2. BASEMENT PLAN, SHOWING DUCT LAYOUT FOR FORCED-AIR SYSTEM IN USE FOR DISTRIBUTING COOL AIR

isolated from the rest of the house by means of the doors opening into the dining room. The entire third story was regarded as an attic and during the daytime was isolated from the rest of the house by means of a door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic during both day and night. With the exception of the space above the northwest bedroom, the third story had hardwood floors laid on pine sub-flooring. In the space above the northwest bedroom one inch of insulating blanket was nailed to the upper edges of the floor joists. Hence, all second floor ceilings were at least equivalent to lath and plaster with flooring above it. No cooking was done in the kitchen, but the heat transmitted through the glass doors from the sun-parlor, which was not ventilated by opening the windows, compensated for this to a certain extent. In all respects

[2] Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 95.

the state of the Residence was comparable for the work done during both summers.

The arrangement of the forced-air heating plant and fan, hereafter referred to as the basement fan, is shown in Fig. 2. For the purpose of this investigation all of the return ducts were blocked unless otherwise specifically stated. For a few tests the outdoor air was taken into the fan through a connection from the nearest basement window. This connection was found to restrict the flow of air, however, and for later tests it was removed, and the fan was allowed to take air directly from the basement, the outdoor air coming in through the open basement door. This fan delivered from 1760 to 2142 cu ft of air per minute.

The arrangement of the attic fan is shown in Fig. 3. This 24-in. fan was installed in the doorway at the head of the stairs leading from the second

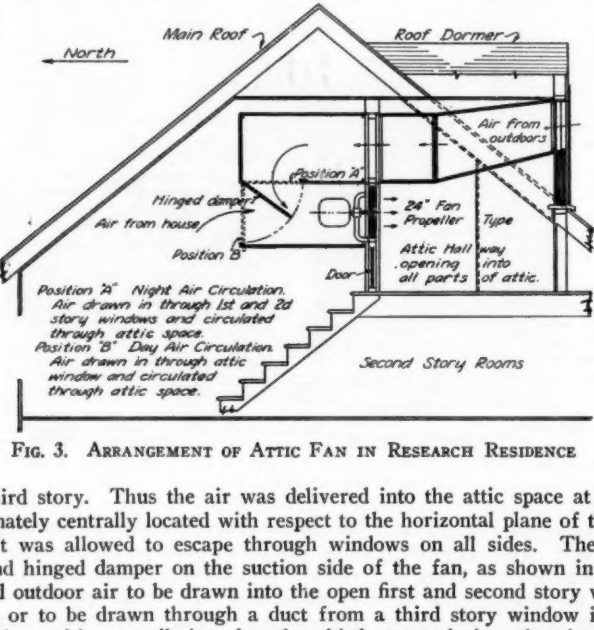


FIG. 3. ARRANGEMENT OF ATTIC FAN IN RESEARCH RESIDENCE

to the third story. Thus the air was delivered into the attic space at a point approximately centrally located with respect to the horizontal plane of the third story. It was allowed to escape through windows on all sides. The use of a box and hinged damper on the suction side of the fan, as shown in Fig. 3, permitted outdoor air to be drawn into the open first and second story windows at night, or to be drawn through a duct from a third story window in order to provide positive ventilation for the third story during the day. This arrangement did not interfere with the full capacity of the fan for circulating the air at night, and the somewhat reduced capacity was more than sufficient for ventilating the attic during the day. The fan delivered approximately 4000 cu ft of air per minute.

METHOD OF CONDUCTING TESTS

During the entire summer from May 23 to September 23, continuous records were made, by means of temperature recorders, of the following air tempera-

tures: outdoor, dining room, kitchen, first story hall, living room, east bed room, southwest bed room, northwest bed room, and third story hall. Other incidental air temperatures were observed at regular intervals. Relative humidities both indoors and outdoors were observed by means of an aspirating psychrometer. The outdoor dry-bulb reading on this psychrometer served as a check on the outdoor temperature read from the recorder chart. Surface temperatures of the north, south, and west walls and of the northwest bed room, east bed room, dining room and living room ceilings were observed by means of thermocouples. Thermocouples were also used to observe the surface temperature of tables in the dining room and east bed room. These data were plotted on a continuous chart, and this chart was used as a basis for the analysis of the results.

Twenty-hour hours operation constituted a test, and for all tests, with the exception of those in *Series 12,* the attic was ventilated by natural circulation through the windows during the day. At night the natural circulation in the attic was employed for all series except those involving the attic fan. In the latter case the fan delivered the air into the attic, and the circulation was outward through the open windows. The Residence was operated strictly on the schedules described in connection with the following enumeration of the different test series:

Series 1. Restricted Natural Circulation.

8 windows, 4 on the first story and 4 on the second, were opened halfway at 9 p. m. and closed at 6 a. m. on the following morning. The attic door was kept closed.

Series 2. Maximum Natural Circulation.

All windows on both stories were opened wide from bottom and the attic door was opened at 6 p. m. and closed at 6 a. m. on the following morning.

Series 3. Restricted Basement Fan Circulation.

8 windows were opened halfway and basement fan started, circulating outdoor air, at 9 p. m. The windows were closed and the fan stopped at 6 a. m. on the following morning. (Comparison series with *Series 1.*) 1760 cfm fan delivery or 7.5 air changes per hour.

Series 4. Basement Air Recirculation.

Basement air was recirculated for short period in early evening hours with house closed up. This was followed by circulation of outdoor air by basement fan with all house windows opened wide (1 window in each of 3 second story rooms was opened from top) and attic door opened. The windows were closed and fan stopped at 6 a. m. on the following morning.

Series 5. Maximum Basement Fan Circulation.

All windows were opened wide (1 window in each of 3 second story rooms was opened from top), attic door was opened and the basement fan was circulating outdoor air at 6 p. m. Windows closed and fan stopped at 6 a. m. on the following morning. (Comparison series with *Series 2.*) 2142 cfm fan delivery or 9.1 air changes per hour.

Series 6. Day Cooling with Ice Plant.

The ice plant was operated in the day time until the outdoor temperature had dropped below the indoor temperature, when night air cooling was started, making use of natural ventilation, the basement fan or the attic fan as in *Series 2, 5,* or *8.*

Series 7. Attic Fan Circulation with Windows Opened Half Way (except halls).

The fan was operated from 6 p. m. to 6 a. m. on the following morning. 3980 cfm air delivery or 16.8 air changes per hour. (Comparison series with *Series 2* and *5.*)

Series 8. Attic Fan Circulation with Windows Opened Wide (except halls).

Same as *Series 7* except that the windows were opened wide. No difference in air delivery.

Series 9. Room Cooling Unit on First Story.

The room cooling unit was operated on first story in day time. Night cooling with the attic fan same as in *Series 7* and *8* except for starting time.

Series 10. Attic Fan Circulation with Only Second Story Windows Open Wide.

The fan was operated from 6 p. m. to 6 a. m. on the following morning. 3980 cfm air delivery or approximately 33.6 air changes per hour on second story.

Series 11. Natural Circulation with Only Second Story Windows Open Wide.

Same as *Series 10* except natural circulation in effect. Attic door open.

Series 12. Attic Fan Circulation in Day Time Through Attic Space Only.

Night circulation same as *Series 8* attic fan circulation.

For the purpose of comparing the fan circulation with natural circulation these series may be divided into three related groups. The first group consists of *Series 1* and *3,* the second of *Series 2, 5, 7* and *8,* and the third of *Series 10* and *11.* Other comparisons, of course, may be made between groups.

When night cooling with outdoor air was used to supplement cooling with ice in the daytime, as in *Series 6,* the cooling plant was adjusted so that the rate of ice meltage could not exceed the equivalent of 700 lb, exclusive of the basement loss, in about 5 hours. After closing the windows and stopping the fan in the morning the indoor temperature was allowed to rise until it reached 81 F, at which time the cooling plant was started. This was allowed to operate at a constant rate until the entire 700 lb of ice were melted unless the indoor temperature decreased below 79 F, or unless the outdoor temperature dropped more than 3 F below the inside temperature. In the case first mentioned, the cooling plant was stopped until the inside temperature again rose to 81 F. In the second case the cooling plant was stopped for the night and the windows were opened and the fan started. Whenever the ice plant was stopped before the time that the windows were opened, the pump was stopped, thus discontinuing the circulation of water through the cooling coil, but the fan was allowed to run in order to maintain recirculation of the air in the house. The 3 F difference between indoor and outdoor temperatures was allowed in order to provide for the difference in relative humidity indoors and

outdoors. The latter difference was such that it required about 3 F lower dry-bulb temperature outdoors in order to have the same effective temperature both indoors and outdoors. If the windows were opened before these two

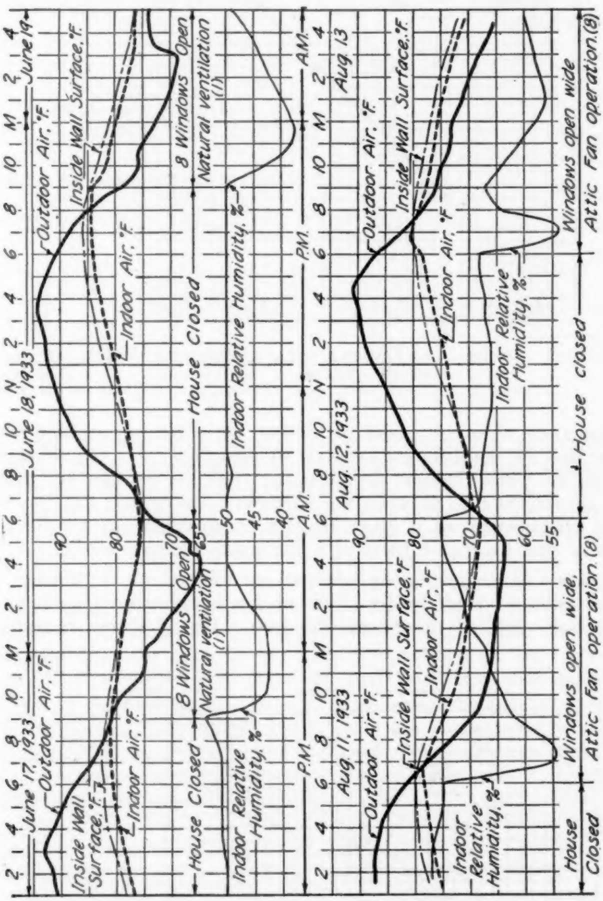


Fig. 4. Daily Temperature Cycles Showing Periods of Night Cooling with Least and Most Favorable Methods of Operation

effective temperatures were equalized it resulted in an undesirable increase in effective temperature indoors.

Results of Tests

Influence of Methods of Operation on Conditions at Night

A study of the results involves the recognition of three separate natural subdivisions or objectives. These are: (1) the study of the conditions at night,

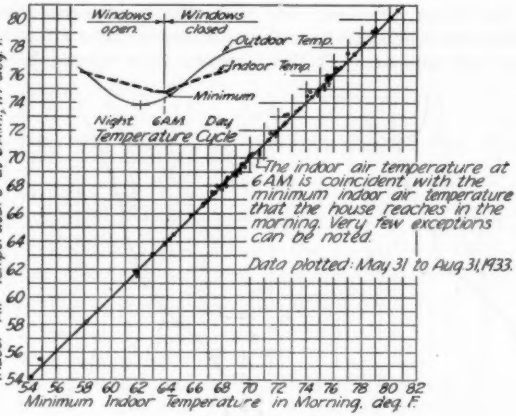


FIG. 5. INDOOR AIR TEMPERATURE AT MINIMUM VALUE AT 6 A.M.

between the times of opening and of closing the windows, as influenced by the weather and by the different methods of operation; (2) the study of the conditions in the house the next day resulting from the weather and from the conditions existing at 6 a. m. when the windows were closed; (3) the prediction of the effectiveness of a given method of operation, or of the cooling required, from the weather reports of a particular summer season.

Table 1 shows the volume of air delivered per minute and the number of air changes per hour produced by the different fan arrangements used, together with the power required to operate the fans. Fig. 4 shows characteristic

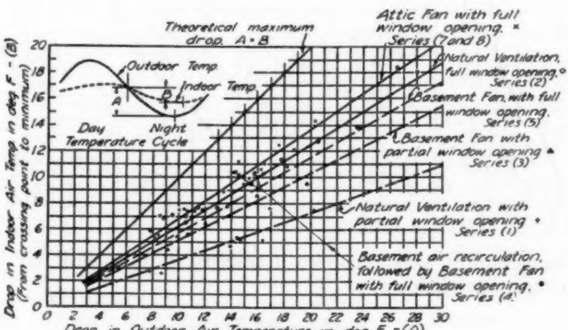


FIG. 6. CURVES OF TEMPERATURE DROPS FOR VARIOUS METHODS OF NIGHT AIR COOLING IN RESEARCH RESIDENCE (SUMMER OF 1933)

TABLE 1. FAN DATA

Test Series	Designation of Fan	Windows	Time of Operation	Air Volume CFM[a]	Air Changes per Hour	Kilowatt-Hours per 12 Hours	Speed RPM
3	Basement	8 windows open	9 p.m. to 6 a.m. Night Air	1760	7.5	4.50	567
4, 5 and 6	Basement	All windows open[c]	6 p.m. to 6 a.m. Night Air	2142	9.1	5.40	567
4 and 6	Basement	Windows closed. Recirculation	Daytime	1482	6.3	4.20	567
7 and 8	Attic	All windows open	6 p.m. to 6 a.m. Night Air	3980	16.8	3.85	845
10	Attic	2nd story windows open only	6 p.m. to 6 a.m. Night Air	3980	33.6[b]	3.85	845

[a] Based on density of air of 0.074 lb per cu ft.
[b] First story windows closed and only second story regarded as active.
[c] Basement air taken through basement door. Inlet duct open to basement.

curves for outdoor temperatures, indoor temperatures and indoor relative humidities for the least and most favorable methods of operation for two similar days. In the first case, shown by the upper curves in Fig. 4, 8 windows were partly opened at 9 p. m. and no fan was used. During the early morning hours of June 18 the outdoor temperature reached a minimum of 65.0 F, while the indoor air temperature did not fall below 75.5 F, or a difference of 10.5 F was shown. During these hours the indoor air temperature remained equal to or slightly above the temperature of the inside surface of the exposed walls, thus permitting no transfer of heat from the walls except by conduction to the outdoor air. The temperature of the air in the house rose to 84.5 F on June 18 following this cycle of operation at night.

In the second case, shown by the lower curves in Fig. 4, the lower sash of all of the windows, which were double hung, was opened to the fullest extent at 6 p. m. and the attic fan was used. The minimum outdoor temperature in the early morning hours of August 12 was 64.0 F, nearly the same as in the first case, but the indoor air temperature dropped to 68.0 F as compared with 75.5 F for the first case. That is, the difference between indoor and outdoor temperatures was 4.0 F as compared with 10.5 F. During the whole night the temperature of the indoor air was distinctly below that of the inside surface of the exposed wall, thus permitting heat to be transferred from the walls to the air which was being removed from the house. The temperature of the air in the house next day rose to only 79.5 F before the windows were opened as compared to 84.5 F for the previous case.

In all cases, the temperature of the surface of the first floor ceilings and of the furniture remained practically the same as that of the indoor air of the room involved. The temperature of the surface of the second floor ceilings was usually from one to three degrees higher than that of the air in the room. (See Fig. 8.)

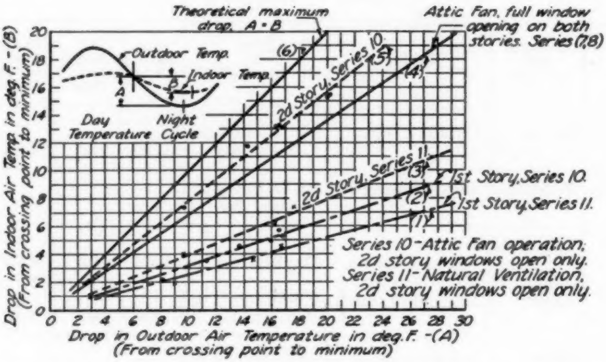


FIG. 7. CURVES OF TEMPERATURE DROPS FOR NIGHT AIR COOLING BY MEANS OF ATTIC FAN. (RESEARCH RESIDENCE DATA, SUMMER OF 1933)

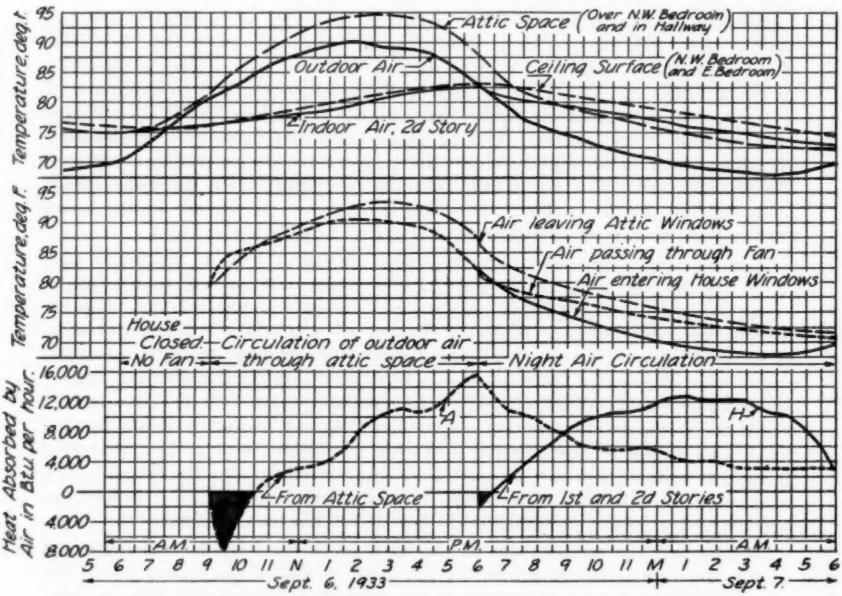


FIG. 8. TEMPERATURE CYCLE OBTAINED WITH ATTIC FAN OPERATION (DAY AND NIGHT)

From Fig. 4 it may be observed that conditions in the house at 6 a.m. when the windows were closed were directly dependent on the weather and the method of operation between the times of opening and closing the windows Since the house was always closed and no fan was operated in the daytime, conditions in the house the next day were determined by the conditions existing at 6 a. m. and by the weather during the day. Inasmuch as these conditions at 6 a. m. were dependent on the method of night operation, the conditions next day were a reflection of the method of operation. But since the conditions at 6 a. m. could be considered as having been established by any independent means, and the resulting conditions during the day would be the same independent of the means required to establish those at 6 a. m., for purposes of analysis of the different methods of night operation the daytime conditions may be regarded as not directly dependent on these methods, and the attention may be concentrated on the part of the cycle included between the times of opening and closing the windows.

The effectiveness of any given method of night operation is directly indicated by the ratio of the rate of decrease in the indoor temperature to the rate of decrease in the outdoor temperature, and the relative effectiveness of the different methods could be compared by comparing these ratios. However, the rates of decrease are also proportional to the drops in temperature from the time that the indoor and outdoor air attain the same temperature, as represented by the crossing point of the two air-temperature curves shown in Fig. 4, to the times at which these two air temperatures each become a minimum. Since these temperature drops were useful in subsequent calculations, they were used as the basis of comparison, instead of the ratios of the rates of temperature decrease. These temperature drops are further useful in that with a known outdoor minimum temperature and a known or assumed temperature at the crossing point, they serve to establish the minimum indoor temperature that would be attained. That this minimum indoor temperature was also the same as the indoor temperature existing at the time that the windows were opened at 6 a. m. is shown by Fig. 5.

A comparison of the temperature drops, *A* and *B,* from the crossing point to the minimum outdoor and indoor temperatures for the different methods of night operation is shown in Fig. 6. The time of the crossing point was not always coincident with the time at which the windows were opened, and the temperature at the crossing point was somewhat influenced by the history between these two times. Hence in order to obtain the effect of the fan alone the comparison must be made between two series for which the windows were opened at the same time and which differed only in that the fan was used for one and not for the other. Other comparisons are valid, however, in that the temperature drop curves reflect the whole history from the time of opening the windows and this whole history is inherent in the particular series under consideration.

From Fig. 6 it is evident that *Series 1* in which 8 windows were opened at 9 p. m. was the least favorable since it gave the least drop in indoor temperature for a given drop in outdoor temperature. The house was stuffy and uncomfortable all night and the temperature was far too high for comfort the next day. A companion series, *Series 3,* run with the same schedule of window opening and with the basement fan delivering 1760 cu ft of air per

minute taken from outdoors, resulted in marked improvement, but did not entirely correct conditions.

From the experience with *Series 1,* it became evident that some advantage was to be gained by opening the windows earlier even if the outdoor tempera-

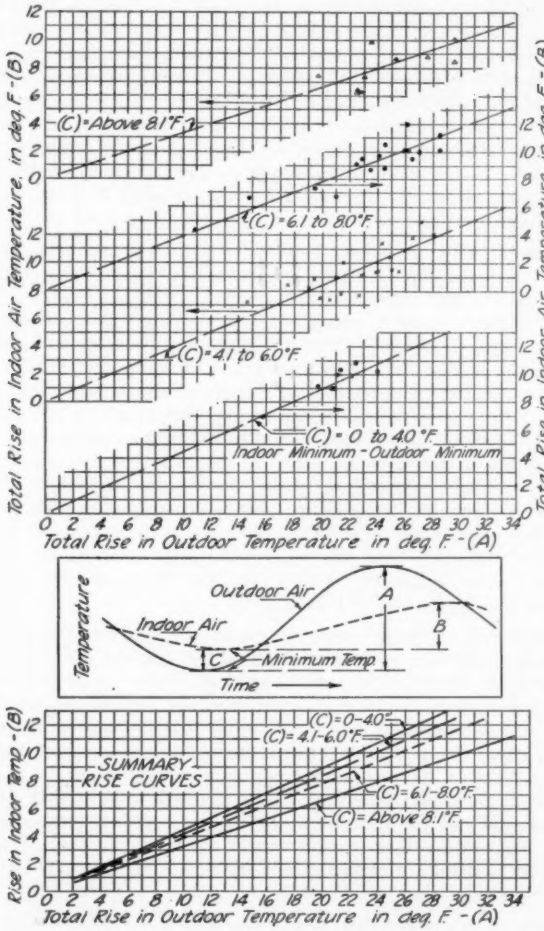


FIG. 9. TEMPERATURE RISE CURVES

ture was somewhat higher than that indoors. *Series 2* was therefore run without a fan, but with all of the windows in the house opened to the fullest

extent possible and in addition the attic door opened in order to obtain the advantage of the full chimney action of the heated air in the house. As shown by Fig. 6, this resulted in very marked improvement over the results from *Series 1,* and in considerable improvement over those from *Series 3.* Conditions in the house were quite comfortable at night and the indoor temperatures the next day were not unreasonable.

It is probable that the Research Residence was better adapted to natural ventilation than the average type of residence would be. The attic was equivalent to a full third story and the dormer windows extended nearly the full

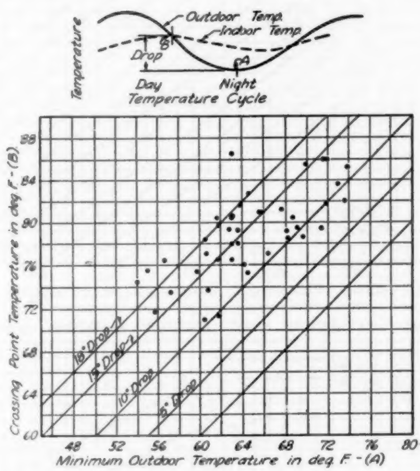


FIG. 10. NORMAL RANGE OF CROSSING TEMPERATURE CORRESPONDING TO MINIMUM OUTDOOR TEMPERATURE. (RESEARCH RESIDENCE DATA, SUMMER OF 1933)

height of the story. A 2.5 ft by 6.5 ft door at the head of the stairs provided ample area for the passage of air into the attic. Hence conditions were particularly favorable for obtaining and utilizing a relatively large chimney effect. In houses with only a small trap door in the second floor ceiling and small attic windows, this chimney effect might be greatly reduced.

Series 5 was run on the same schedule of window opening as *Series 2* but with the basement fan delivering 2142 cu ft of air per minute taken from outdoors. The results from this arrangement were not quite as favorable as those obtained with *Series 2* for which no fan was used. This can not be explained on the basis that favorable wind movement aided the natural ventilation in *Series 2.* The different points on the curves represented days covering the whole range of wind movement at night and sun effect by day. During most of the season the nights were comparatively calm, and the wind velocity

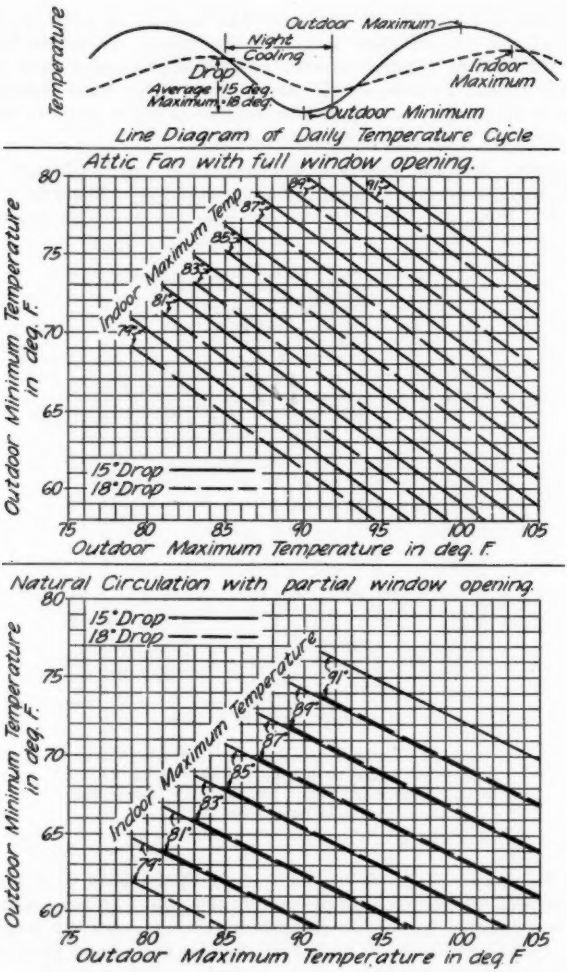


FIG. 11. CALCULATED MAXIMUM INDOOR AIR TEMPERATURES (RESEARCH RESIDENCE DATA, SUMMER OF 1933)

varied from 2 to 7 mph. Except for an occasional storm the days had a high percentage of sunshine. Hence, the most probable explanation appears to be that more air was circulated through the house as a whole by full natural ventilation than by the basement fan. It is also possible that in the latter case some of the air short-circuited out of the windows, although no positive

evidence could be obtained on this point owing to the low velocity at the windows.

In *Series 4* an attempt was made to take advantage of the reservoir of cooler air existing in the basement by recirculating this basement air through the house for about one hour before opening the windows. This series is

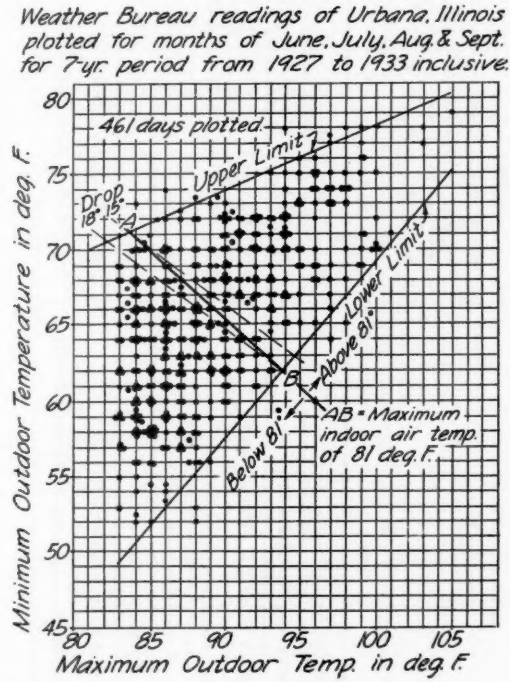


FIG. 12. COMBINATIONS OF OUTDOOR MAXIMUM AND MINIMUM TEMPERATURES IN URBANA, ILLINOIS

comparable with *Series 5* and the two points representing these tests are shown in Fig. 6. This method of operation represented a slight gain over the results from *Series 5,* but the temperature of the basement air soon rose to practically the same as that upstairs and the amount of gain can not be regarded as sufficient to offset the complication in the operating routine. Furthermore, this method of operation appeared to accentuate odors. With all of the different test series, there was a tendency for odors to become slightly noticeable in the afternoon, particularly on the second story, after the windows were closed in the morning, but not to the extent of becoming objectionable.

The most favorable results shown in Fig. 6 were obtained from *Series 7* and *8* in which the attic fan, drawing 3980 cu ft of air per minute into the

TABLE 2. RESULTS OF OBSERVATIONS WITH A KATA-THERMOMETER

Series	Date 1933	Time of Test	Operating Notes	Air Temperature (Average) Deg F				Air Velocity, FPM		
				First Story		Second Story		Dry Kata		
				D. B.	W. B.	D. B.	W. B.	First Story Rooms	First Story Hall	Second Story Rooms
1	2	3	4	5	6	7	8	9	10	11
Still Air	6–24	2–4 p.m.	House closed up; No fan	79.2	67.9	82.0	68.5	4.3	4.6	3.0
2	8–22	7–9 p.m.	Natural ventilation; Maximum effect	75.5	63.4	77.8	64.3	3.2	3.7	2.8
4	6–24	4–5 p.m.	Recirculation of house air. House closed	79.6	68.4	81.5	69.1	19.4	—	6.6
5	6–26	7–9 p.m.	Basement fan maximum circulation	81.2	73.3	82.2	73.7	16.7	5.2	10.9
7	7–17	7–9 p.m.	Attic fan. All windows open halfway	74.9	61.3	74.2	61.0	12.2	25.8	4.2
8	7–30	7–9 p.m.	Attic fan. All windows open wide	83.1	71.5	83.9	71.7	19.2	30.7	6.9
10	9–2	8–9 p.m.	Attic fan. 2nd story windows open only	72.5	67.9	71.7	68.9	—	4.9	17.5
11	9–18	7–8 p.m.	Natural ventilation. 2nd story windows open only	73.0	63.6	74.9	63.0	—	3.9	2.9

NOTES: Columns 5 to 8 inclusive, for dry-bulb and wet-bulb temperatures, are the average of 4 readings in each case.

Columns 9, 12, and 15 are the average of Kitchen, Dining Room, and Living Room.

Columns 10, 13, and 16 are the values for first story hall only.

Columns 11, 14, and 17 are the average of E. Bedroom, S. W. Bedroom, N. W. Bedroom, and Bathroom.

TABLE 2 (CONTINUED)

Series	Heat Loss, H, Btu. per Sq. Ft. per Hr. — Dry Kata: First Story Rooms	Dry Kata: First Story Hall	Dry Kata: Second Story Rooms	Wet Kata: First Story Rooms	Wet Kata: First Story Hall	Wet Kata: Second Story Rooms	Weather and Wind Conditions
	12	13	14	15	16	17	18
Still Air	35.5	36.1	28.7	133	143	118	Clear. Very hot; Max. = 96 F. West—13 mph
2	42.6	49.6	37.6	167	184	158	Cloudless skies. Cooling slowly; Temp. = 74 F. Practically calm. NE-4
4	40.9	—	32.6	161	—	138	Same as (a)
5	39.3	33.5	33.3	141	119	135	Partly cloudy. Wind SW-2
7	48.8	56.3	44.9	178	232	168	Almost cloudless. West wind mild
8	29.8	39.4	27.4	152	182	137	Almost cloudless. Calm. SW-3
10	—	48.9	59.8	—	143	174	Very humid. Temp. = 70 F. Calm
11	—	46.6	42.2	—	162	160	Cloudless. Outdoor Temp. = 71 F. Calm

first and second stories, was used. No difference was observed between *Series 7* for which the lower sash were raised half way, and *Series 8* for

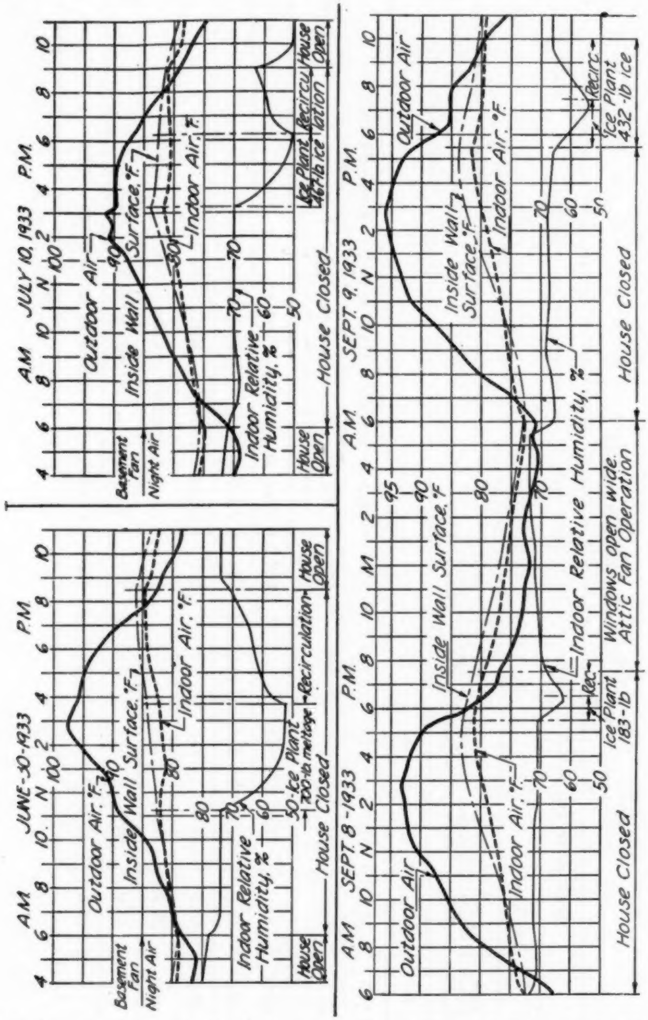


FIG. 13. DAILY TEMPERATURE CYCLES SHOWING PERIODS OF OPERATION OF COOLING PLANT

which they were raised all of the way. From Fig. 6 it is apparent that there was not a great amount of difference between the three most favorable methods of operation, and that even with the 16.8 air changes per hour produced by the attic fan the indoor temperature was not reduced to the same temperature

as the outdoor air. The latter condition would be represented by the line designated as the theoretical maximum drop. The results of the tests indicate that considerable benefit may be obtained from the use of an attic fan drawing a generous amount of air into the house through the open windows at night. In a house similar to the Research Residence, having two full stories and an attic having dormer windows and a large attic door, practically as much benefit may be obtained without any fan by opening all of the windows and the door leading into the attic. In case it is not desirable to open a large proportion of the windows at night considerable benefit may be obtained from the use of either a fan in the attic or one installed in connection with a forced air heating system and drawing air from the outdoors. The power required, as shown by Table 1, ranged from 3.85 to 5.40 kw-hr for 12 hours of operation at night.

Series 10 and *11* were run with the attic fan and with full natural ventilation respectively, with the windows opened on the second story only. In this case all of the 3980 cu ft of air per minute was drawn in through the second story windows, giving 33.6 air changes per hr based on the cubic contents of the second story. The resulting temperature drops for the first and second stories are shown in Fig. 7. Comparing curve *No. 3* in Fig. 7 with the corresponding curve in Fig. 6, it is evident that the natural ventilation with full window opening was not as effective for the second story alone as it was when all of the windows on both stories were opened. Curves *Nos. 3* and *5* show that when the fan was used in connection with the second story alone there was more marked improvement than there was when it was used in connection with both stories. This is indicated by the fact that the distance between curves *Nos. 3* and *5* is greater than that between the corresponding curves in Fig. 6, and also by the fact that curve *No. 5* lies above curve *No. 4* which has been transferred to Fig. 7 from Fig. 6. Considering curve *No. 1* in Fig. 7, it is also evident that natural ventilation on the second story alone resulted in a very small temperature drop on the first story as compared with the average for the house shown in Fig. 6 resulting from natural ventilation on both stories, and while the use of the fan as shown in curve *No. 2* resulted in some improvement the drop was still not comparable with that occurring when either the fan or full natural ventilation was used in connection with both stories.

In order to determine whether there was any difference in cooling effect produced by the air movement resulting from the different methods of night operation Kata-thermometer readings were made at the centers of the rooms. The results are shown in Table 2. The criterion for comfort as stated by Dr. Leonard Hill is that the cooling effect for the dry Kata should exceed 40 and for the wet Kata 135 Btu per sq ft per hr. Comparing the results from *Series 2* and *7* it appears that the use of the attic fan increased the air movement from a velocity of 3.2 to 12.2 ft per minute on the first story and from 2.8 to 4.2 ft per minute on the second. Since the wet- and dry-bulb temperatures were comparable for these two cases the Kata cooling effects are also comparable. The dry Kata cooling effect was increased from 42.6 to 48.8 Btu per sq ft per hr on the first story and from 37.6 to 44.9 on the second, and the wet Kata cooling effect was increased from 167 to 178 on the first story and from 158 to 168 on the second. All of these cooling effects were greater than the 40 and 135 Btu per sq ft per hr required for comfort.

Comparing the results from *Series 7* and *8* it may be noted that when the attic fan was used the velocities at the center of the room were increased somewhat when the windows were opened wide instead of only half way. Since the dry-bulb temperatures were not the same the Kata cooling effects are not comparable. Since the first story hall acted as a duct to convey the air from the first story, the velocity in the hall was considerably higher than that in the first and second story rooms.

Comparing the averages for the first and second stories in *Series 5* with those in *Series 7* and *8* it may be noted that there was some tendency for the basement fan to cause greater disturbance of the air in the rooms than there was for the attic fan. The velocities determined by the Kata thermometer, excluding the lower hall which acted as a duct when the attic fan was used, were higher for the basement fan than for the attic fan. In the former case the air was delivered into the room as a jet from the register, while in the latter it was drawn more uniformly into the room through the windows. Hence more disturbance would result in the former case than in the latter.

Comparing *Series 10* and *11* it is evident that with the windows opened on the second story only, the air movement in the second story rooms was increased from 2.9 ft per minute when no fan was used to 17.5 ft per minute when the attic fan was used. Also this velocity was much greater than the 4.2 and 6.9 ft per minute obtained on the second story when the attic fan was used and all windows on both stories opened. A study of the velocities at the open windows in the latter case proved that the air was being drawn into the first story windows at an average velocity of 57.9 ft per minute and into the second story windows at an average velocity of 46.4 ft per minute. This indicated that when the attic fan was used in connection with both stories there was no tendency for the air to short-circuit through the second story windows at the expense of the first story, but, on the contrary, the suction resulting from the chimney action of the house was greater on the first floor than on the second and adding to the effect of the suction produced by the fan caused a greater flow of air into the first story windows than into the second.

A study was made to determine whether any advantage resulted from recirculating the air in the house by means of the basement fan while the house was closed during the daytime. These results are shown in Table 2, for still air and for *Series 4*, under date of June 24. When no recirculation occurred the air velocities at the center of the rooms were 4.3 and 3.0 ft per minute on the first and second stories respectively. The corresponding dry and wet Kata cooling effects were 35.5 and 28.7, and 133 and 118 Btu per sq ft per hr. These cooling effects were below the threshold for comfort. Immediately after making these observations the basement fan was started in order to recirculate the air. The velocity at the center of the rooms was increased to 19.4 and 6.6 ft per minute on first and second stories respectively and the corresponding dry and wet Kata cooling effects were increased to 40.9 and 32.6, and 161 and 138. The latter are very close to, or just above the threshold for comfort. Hence, while the effect was not very marked, some advantage resulted from recirculating approximately 1482 cu ft of air per minute, and, in border-line cases, this procedure might convert an uncomfortable atmosphere into one that is just comfortable.

A few tests, designated as *Series 12,* were run with the attic ventilated during the day by means of the attic fan instead of by natural ventilation through the windows. The temperature of the air in the attic together with the temperature of the surface of the ceiling and the air in the rooms of the second story for a characteristic day, are shown in Fig. 8. The temperatures of the air entering the fan and leaving the attic windows are also shown in this figure. These temperatures taken in connection with the volume of air delivered by the fan afforded a means of calculating the heat absorbed either from the attic or second story alone, or from both.

Considering the attic alone, during the day the air from outdoors entered the fan through the duct shown in Fig. 3 and was forced out through the windows. Thus the temperature rise was representative of the heat absorbed from the attic. A comparison of the power required by the fan motor for day and night operation indicated that no appreciable reduction in fan capacity was caused by the duct arrangement used during the day. Hence the full fan capacity of 3980 cfm was used in the calculations for curve *A* shown in Fig. 8. This curve gives the heat absorbed from the attic and shows a maximum rate of 15,600 Btu per hr, or the equivalent of 1.3 tons of refrigeration. However, a comparison of the results from two similar days, one with the fan in operation and one without, showed no appreciable lowering of the temperature of the air in the attic resulting from the use of the fan, and no appreciable difference in either the temperature of the surface of the ceiling or of the air on the second story for the two cases. The temperature of the air in the attic was measured at two points, approximately 4 ft above the floor. Hence it is probable that the temperature of the air in the peak of the roof and near the attic ceilings when no fan was used was much warmer than the air nearer the floor where the measurements were made. This formed a stationary pool of hot air above the level of the windows that served to reduce the heat transmission through the roof and to insulate the air nearer the floor. When the fan was used this pool was swept out and the movement of the air over the upper surfaces increased the heat transmission. Thus a large quantity of heat was removed and carried out by the air going through the windows, but the greater part of this represented heat that did not penetrate the lower stratum of air when the fan was not used. If this were the case, the measured temperature of the air near the floor would remain about the same whether the fan was or was not used. Thus the heat transmitted through the floor would remain about constant for the two cases and no change would occur in the temperature of the surface of the ceiling or of the air on the second story. Thus, while the fan apparently removed a large quantity of heat from the attic during the day, no appreciable improvement in conditions on the second story resulted as compared with conditions existing when the attic was ventilated by means of open windows alone. More refined methods of testing might have indicated a slight difference, but if the difference was sufficiently small to demand more refined testing methods, the cost of operating the attic fan during the day could not be justified under the conditions existing at the Research Residence. It is possible that in a poorly ventilated attic, with no windows, or with small windows in positions not adapted to cross ventilation, the use of an attic fan during the day could be entirely justified.

The rate at which heat was removed from the first and second stories by

the operation of the attic fan at night is shown by the curve *H* in Fig. 8. In this case the air was drawn into the windows from outdoors and all of the air passing through the house was delivered by the fan. Hence the difference in temperature between the air entering the fan and that outdoors was representative of the heat absorbed from the first and second stories. It may be noted that this heat amounted to approximately 12,000 Btu per hr or the equivalent of one ton of refrigeration, over a period of about 3 hr. Since approximately the same amount of heat loss would occur by conduction to the outdoors through the exposed walls, irrespective of whether night air was or was not circulated, the amount of heat represented by curve *H* represents in some measure the gain resulting from the circulation of air from outdoors at night. Owing to the presence of cross currents when full natural ventilation without the fan was used it was not possible to measure the actual heat absorbed in this case. However, the evidence presented in Fig. 6 indicates that very nearly as much heat was removed by the full open window ventilation in *Series 2* as that removed by the attic fan in *Series 7* and *8*. The portion of the curve *A* corresponding to curve *H* in Fig. 8, represents the additional heat removed at night from the attic alone. The whole of this amount can not be regarded as a gain for the next day, however, because due to the high heat transmission of the roof and to the fact that the windows in the attic were not closed during the day, the gain could not be conserved, as it was for the first and second stories on which the windows were closed at 6 a. m.

Effect of Methods of Night Operation on Condition Next Day

Following a given method of night operation, the maximum temperature attained the next day after the windows have been closed at 6 a. m., depends upon the indoor temperature attained at 6 a. m. and the subsequent rise in outdoor temperature. As indicated by Fig. 5 this indoor temperature at 6 a. m. was the same as the minimum temperature attained indoors. Hence, it was determined by the night history, or the temperature drop shown in Fig. 6 corresponding to the particular method of night operation under consideration. The difference between the minimum outdoor temperature and the indoor temperature at 6 a. m. may be represented by *C*, shown in the diagram in Fig. 9. The observed data from all of the tests not involving artificial cooling have been classified under four arbitrary values of the temperature difference, *C*, and the results have been plotted in Fig. 9, giving the relation between the rise in indoor temperature and that in outdoor temperature for the different values of *C*.

If the temperature corresponding to the crossing point of the indoor and outdoor temperature curves in the early evening hours is known for a given day, and the subsequent minimum outdoor temperature is obtained from the weather reports, both the indoor temperature drop and the difference, *C*, can be evaluated from the temperature drop curve in Fig. 6 corresponding to the particular method of night operation. The indoor temperature at 6 a. m. can then be found by subtracting the indoor drop from the temperature at the crossing point. By the aid of these data and the rise in outdoor temperature, *A* in Fig. 9, the rise in indoor temperature and the maximum indoor temperature attained in the house next day can be calculated. If some relation could be shown between the temperature at the crossing point and the minimum out-

door temperature following it, or the drop from the crossing point to the minimum outdoor temperature, then, for a given method of night operation, the probable maximum temperature attained in the house next day could be predicted for various combinations of outdoor minimums and succeeding outdoor maximums obtained from weather reports.

The relation between the temperature at the crossing point and the minimum outdoor temperature obtained from the observed data is shown in Fig. 10. Lines representing equal temperature drops from the temperature at the crossing point to the minimum outdoor temperature have been added. From the spread of points in Fig. 10, it may be observed that a number of different crossing point temperatures may exist for any given minimum outdoor temperature. Under these circumstances, for a given series of tests, in which the temperature of the crossing point is known from the test data, the solution for the maximum indoor temperature for the next day is obvious and simple. The difficulty in applying the curves to the prediction of the maximum house temperature in general, when no test is available but when the method of operation is given and the minimum and maximum outdoor temperatures are known from the weather reports, lies in the fact that, while all of the other factors are known or determinable from the curves, the temperature at the crossing point is a casual factor depending on the previous history of the house and the weather. It may be noted from Fig. 10, however, that over the whole range of observed weather conditions, the temperature drops from the crossing point to the minimum outdoor temperature, with few exceptions, were above 10 F and did not exceed 18 F. Hence, it is possible to establish upper limits for the maximum indoor temperature, above which this temperature would not rise with given combinations of maximum and minimum outdoor temperatures.

Assuming 18 F and 15 F drops, from Fig. 10, as characteristic of the most severe and of average weather respectively, the curves in Fig. 11 have been drawn representing maximum probable indoor temperatures resulting from various combinations of minimum and maximum outdoor temperatures occurring in connection with the most and least favorable methods of night operation. These may be used to predict the probable maximum indoor temperature from the weather reports of a given summer season.

Experience at the Research Residence seemed to indicate that approximately 82 F dry-bulb was a critical temperature from the standpoint of comfort. Without artificial cooling, and with the usual indoor relative humidity of from 50 to 60 per cent, conditions were reasonably comfortable if the indoor temperature did not exceed 81 F. This was equivalent to an effective temperature of between 74 and 75.5 F. Hence 81 F dry-bulb was accepted as the upper limit for comfort without cooling, and, for the cooling tests, the cooling plant was started when the indoor temperature reached 81 F. Under these conditions, owing to the reduction in indoor relative humidity when the cooling plant was in operation, the effective temperature remained somewhat below the 75 F indicated as the upper limit on the comfort chart.

Fig. 12 shows the various combinations of maximum and minimum outdoor temperature occurring at Urbana, Illinois, during the past seven summers. Practically all of the plotted data could be included between two lines representing upper and lower limits. From these curves it may be observed that with a maximum outdoor temperature of 100 F the preceding minimum would

not have exceeded 78 F. Judging from the points shown in Fig. 10, with this minimum outdoor temperature, the greatest drop from the crossing point to the minimum would be of the order of 12 F and certainly would not exceed 15 F. Hence for a day on which the maximum outdoor temperature was 100 F, with 78 F minimum outdoor temperature the night preceding and a 15 F drop in temperature from the crossing point to the minimum outdoor, a maximum indoor temperature not to exceed 92 F would be expected if the attic fan had been operated during the night. On the other hand if the house was operated on natural ventilation with partly open windows as in *Series 1,* an indoor temperature of 94.5 might be obtained. During one season comparatively few 100 F days would occur. There might be a considerable number of 95 F days, however. In this case, Fig. 12 indicates a possibility of a minimum outdoor temperature of 76 F and Fig. 10 of a 15 F temperature drop. These conditions would result in a maximum indoor temperature of 88.5 F succeeding a night of operation with the attic fan, and of 92 F for natural ventilation with partly open windows. Both of these indoor maximum temperatures are considerably above the 81 F representing the limit for comfort.

It is of some interest to consider the number of days that the maximum indoor temperature might have exceeded 81 F if the attic fan had been operated at night over the whole period of a season. Assuming from the evidence of Fig. 10 that 15 F and 18 F temperature drops represented limiting conditions for outdoor minimum temperatures above and below 68 F respectively, and obtaining from Fig. 11 the maximum indoor temperature for the actual maximum and minimum outdoor temperatures shown by the weather reports for each day in the summers of 1932 and 1933, it was found that there would have been 37 days in 1932 and 47 days in 1933 on which the maximum indoor temperature might have exceeded 81 F. This, of course, represents the upper limit and it is possible, since Fig. 10 shows that about ⅓ of the total number of observed points were below the 15-deg drop line, that actual operation might have resulted in approximately 25 and 32 days.

The 15 F and 18 F temperature drop lines representing indoor maximum temperature of 81 F have been transferred from Fig. 11 and shown in Fig. 12. Since the 15 F drop line is the most probable upper limit for outdoor minimum temperatures above 68 F and the 18 F for outdoor minimum temperatures below 68 F, the line *AB* represents a limit line above and to the right of which any combinations of outdoor maximum and minimum temperatures may certainly be expected to result in indoor maximum temperatures exceeding 81 F, and below and to the left of which represents such combinations resulting in maximum indoor temperatures not exceeding 81 F. From the points of intersection, *A* and *B,* it is evident that the maximum indoor temperature will always exceed 81 F if the outdoor maximum reaches 94 F, and may do so if the outdoor maximum rises above 84 F. Hence it is probable that night operation with a fan, even under the most favorable conditions, can not be depended upon to result in comfort over the whole of a summer season in a climate similar to that of Urbana, Illinois, unless supplemented part of the time by some form of artificial cooling. It can be used, however, to alleviate conditions even in the most severe weather, and in this respect may prove satisfactory to a considerable number of householders to whom the cost of a cooling plant might be prohibitive.

Circulation of Outdoor Air at Night as a Supplement to Ice Cooling

Only the three most favorable methods of night operation involving the use of the basement and attic fans, and full natural ventilation, were used to supplement ice cooling during the day. Fig. 13 shows temperature and indoor relative humidity curves for somewhat similar days and for two characteristic conditions in the operation of the cooling plant. The test of June 30 illustrates a case in which it was necessary to increase the rate of ice meltage and for which the 700 lb allotment was all melted before the outdoor temperature dropped 3 F below the indoor temperature, thus permitting the windows to be opened. In the case of all of the cooling tests the windows were not opened at any stated time, but the time was determined by the rate of decrease in the outdoor temperature. Only a few days similar to June 30 were encountered for which it would have been necessary to increase the meltage to somewhat more than 700 lb in order to maintain the indoor temperature below 82 F. In these cases the sacrifice in comfort was slight because the indoor relative humidity did not increase rapidly after stopping the ice meltage and the period before the windows were opened was comparatively short. The remaining tests shown in Fig. 13 illustrate cases in which the 700 lb of ice were more than sufficient to maintain 81 F until the outdoor temperature was 3 F below that indoors.

The four tests shown in Fig. 13 prove that the amount of ice used was dependent both on the maximum temperature attained during the day and on the history of the preceding night. The tests of September 8 and 9 indicate that as a hot wave progressed and the maximum and minimum outdoor temperatures increased from day to day, the ice required increased because the cooling plant either had to be started earlier, or had to be operated longer before the outdoor temperature dropped 3 F below the indoor. The latter case is illustrated by the test of September 9 for which the outdoor temperature did not decrease as rapidly as it did on September 8. No cooling was required on September 7, although the maximum outdoor temperature was 91 F, which was comparable with July 10, and very little cooling was required on September 8 when the maximum outdoor temperature rose to 93 F. July 10 was preceded by a hot day during the night of which the minimum outdoor temperature was 69 F as compared with 67.5 F attained in the early morning hours of September 8. Hence the minimum indoor temperature was reduced to only 75 F to start the day of July 10 as compared with 72.5 to start the day of September 8. Again on September 9, the minimum indoor temperature was reduced to 72.5 F in the early morning hours, and September 9, for which the maximum outdoor temperature reached 96 F, required about the same amount of ice as July 10, for which the maximum was 90 F.

A few tests, *Series 9,* with a room cooling unit in the living room proved that, with outdoor temperatures as high as 93 F, the temperature in the living room, lower hall, and dining room could be maintained at from 77 to 80 F. The temperature in the kitchen rose to 82 F but conditions were not particularly uncomfortable because the relative humidity maintained in the kitchen was the same as that in the other first story rooms. The relative humidity was reduced from a value of about 70 per cent to one of about 55 per cent. The unit produced no noticeable effect on the conditions on the second story.

TABLE 3. ICE REQUIREMENTS FOR SUMMERS OF 1932 AND 1933

Series No.	Method of Operation at Night	Summer of 1932					
		Estimated No. of Days Requiring Cooling[b]					Ice Requirement
		June	July	Aug.	Sept.	Total	Tons[a]
1	Natural ventilation. 8 windows partly open at 9 a.m.	21	24	18	2	65	23
3	Basement fan. 8 windows partly open at 9 p.m.	11	18	11	0	40	14
2	Natural ventilation. All windows open at 6 p.m.	9	17	8	0	34	12
7 and 8	Attic fan. All windows open at 6 p.m.	7	15	7	0	29	10

In the case previously discussed, where no cooling was employed, the temperature corresponding to the crossing point of the indoor and outdoor temperature curves was a more or less casual factor, and the prediction of the maximum indoor temperature was confined to certain limiting values. In the case where the cooling plant is to be used consistently throughout the season, however, and the plant is started whenever the indoor temperature rises to 81 F, the temperature at the crossing point could usually be maintained at approximately 80 F. Hence, for a given method of operation at night, the curves in Figs. 6 and 9 used in connection with the minimum and maximum outdoor temperatures obtained from the weather reports for a given season afford a comparatively certain means of predicting the number of days on which the indoor temperature would reach 81 F and necessitate starting the cooling plant. If the ice meltage is limited to 700 lb a day and it is assumed that the 700 lb are always melted, then the product of 700 and the number of days requiring the use of the cooling plant represents the maximum limit of the amount of ice required per season. It is true that a few days were observed for which somewhat more than 700 lb of ice would have been required to maintain the indoor temperature at 81 F, but these days were more than offset by the ones not requiring the entire 700 lb. Hence, the amount of ice obtained by the method outlined would be the maximum.

On the assumption that the attic fan is used at night and the crossing point temperature is limited to 80 F, if an outdoor minimum temperature of 68 F is observed the indoor temperature drop of 8.1 F may be obtained from Fig. 6. This represents an indoor minimum, or temperature at 6 a. m. of 71.9 F. The temperature difference, C, shown in Fig. 9, would then be 3.9 F. If the outdoor temperature attained a maximum of 95 F the next day, the outdoor rise would be 27.0 F, and the indoor rise corresponding to a value of 3.9 F for C would be 12.0 F. Hence, the indoor temperature would rise to 83.9 F if no cooling were employed, indicating that the cooling plant would have to be started. By following this procedure for all of the recorded outdoor maximum and minimum temperatures for the summer the number of days requiring cooling

TABLE 3. *Continued*

Series No.	Method of Operation at Night	Summer of 1933					
		Estimated No. of Days Requiring Cooling[b]					Ice Requirement
		June	July	Aug.	Sept.	Total	Tons[a]
1	Natural ventilation. 8 windows partly open at 9 p.m.	25	24	12	12	73	26
3	Basement fan. 8 windows partly open at 9 p.m.	21	18	6	8	53	19
2	Natural ventilation. All windows open at 6 p.m.	20	15	3	6	44	15
7 and 8	Attic fan. All windows open at 6 p.m.	19	14	3	6	42	15

[a] Based on an ice consumption of 700 lb per day.
[b] Based on indoor temperature exceeding 81 F at some time during the day.

and the maximum amount of ice required for the season may be obtained. This procedure has been followed for four different methods of night operation for the seasons of 1932 and 1933, and the summary is given in Table 3. Neither season would have required any cooling in the month of May.

During the summer of 1932, which included 1538 degree-hours above 85 F, 43 tons of ice were required under the system of operation followed for that summer. It is of considerable interest to note that if advantage had been taken of the possibility of circulating night air and an attic fan had been used under conditions corresponding to those for *Series 7* and *8,* the probable ice requirement would have been only 10 tons. With natural ventilation with all windows open the requirement would have been 12 tons, and with the basement fan with 8 windows partly open it would have been 14 tons. Even with the least favorable method of night operation the requirement would have been only 23 tons. The summer of 1933 included 2310 degree-hours above 85 F, and, as indicated in Table 3, the ice requirements would have been somewhat greater than those for the summer of 1932. However, the table presents concrete evidence that even the least favorable method of circulating the cool air from outdoors at night, when used to supplement some form of artificial cooling during the day, has merit as a means of reducing the cost of summer cooling to an amount that may not be prohibitive for the average householder.

CONCLUSIONS

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted:

1. The circulation of air from the outdoors at night, when used as a supplement to artificial cooling during the day, has considerable merit in reducing the seasonal cooling load that would otherwise be required.

2. The circulation of air from the outdoors at night may make the use of artificial cooling unnecessary for a considerable portion of the summer season.

3. If the best means of circulating air from the outdoors at night had been used at the Research Residence during the summer of 1932 the ice meltage could probably have been reduced from the 43 tons actually used to an amount of the order of 10 tons.

4. The practice of partly opening a few windows at night is not very effective as a means of circulating air from the outdoors.

5. The use of a fan in a forced air heating system to circulate from 6 to 9 air changes per hour from the outdoor at night is much more effective than opening a few windows even when the same amount of window opening is retained for the two cases.

6. The use of a fan in a forced air heating system to circulate 9 air changes per hour is more effective if all of the windows and the attic door or hatchway is opened than if only a few windows are opened.

7. The most effective method of circulating air from the outdoors at night is to open all of the windows and to use an attic fan drawing the equivalent of approximately 17 air changes per hour into the windows of the first and second stories and discharging it into the attic to escape from the attic windows.

8. In the case of a two story house similar to the Research Residence having an ample attic with dormer windows and large attic door, opening all of the windows and the attic door is nearly as effective for circulating air from the outdoors at night as the use of an attic fan producing approximately 17 air changes per hour.

9. There is some advantage in opening the windows at 6 p. m. rather than at 9 p. m. even if the outdoor temperature is slightly higher than the indoor temperature at 6 p. m.

DISCUSSION

A. P. KRATZ (WRITTEN): It is of some interest to determine the limiting number of air changes that would be required to prove effective for night cooling in a house of the type of the Research Residence. The results presented in the paper may be extended to include this determination.

The slope of the curves in Figs. 6 and 7 are a measure of the relative effectiveness of the different methods employed. Furthermore, the relative effectiveness of the different methods may be considered as dependent on the amount of air circulated and not inherent in the particular method for circulating the air.

The slopes of a number of the curves from Figs. 6 and 7 are shown in Fig. A plotted against the corresponding amounts of air circulated, expressed as number of air changes per hour. A slope of 1.0 would represent the maximum effectiveness, inasmuch as it would represent conditions under which the indoor temperature would be reduced to the same value as that outdoors. From Fig. A it may be observed that with the smaller numbers of air changes the slope increases very rapidly as the number of air changes increases, but that the rate of increase in the slope becomes much less for the larger numbers of air changes. The slope curve becomes asymptotic to a line representing a slope of 1.0 at a value certainly in excess of 50 air changes per hour.

It is therefore evident that night cooling does not become reasonably effective until the number of air changes per hour reaches a value of approximately 9, and that

above a value of 30 air changes per hour very little gain results from increasing the amount of air circulated. While the use of 30 air changes per hour is practical, there is some doubt as to whether the increased cost of current could be justified in increasing the number of air changes per hour from 20 to 30.

G. B. HELMRICH (WRITTEN): During this past summer the ice cooling installation installed in a Detroit residence, under the sponsorship of The Detroit Edison Co., was operated in a manner quite similar to that used at the Research Residence, and a comparison of the results obtained in the two residences should prove of value.

The Detroit residence is about the same size as that of the Research Residence and is of a similar design; the only real difference being that the former is unusually well insulated. The cooled air was circulated by means of a furnace fan which has a capacity sufficient to provide about 7 air changes per hour for both stories. As this residence is not equipped with an attic fan, cooling by outdoor air was accomplished by using the furnace fan to draw in outdoor air through an open basement window. Artificial cooling was restricted by limiting the ice quantities to from 500 to 700 lb per day, and any additional cooling required was accomplished by circulating outdoor air whenever temperatures were such that it could be done to advantage. The ice charge lasted for periods varying from 3 to 5 hours, following which the water pump was shut down and the air was recirculated until such time as the temperature of the outdoor air had fallen to a temperature equal to that indoors. This usually occurred at around 7:30 to 8:30 p.m., and from this hour until about 6:30 the following morning, outdoor air was circulated through the house. Although the temperature records do not indicate that night air cooling had any appreciable effect on the need for artificial cooling the following day, the use of the fan did make it possible to stop the artificial cooling at about 6:00 to 7:00 p.m. In addition, the occupants of the house reported an improvement in comfort in the downstairs rooms, during the early evening hours, due to the movement of the air; a result not unexpected, as temperature records alone do not indicate the entire cooling effect produced.

If unrestricted cooling had been done last summer, as it was in the summer of 1932, the equivalent ice consumption would have been of the order of 18 tons; the actual equivalent ice consumption last summer was about 6 tons. This 66 per cent reduction in ice consumption checks very closely with the saving made at the Research Residence, when operating the cooling system in a manner similar to that used in the Detroit residence, and shown by the author in his Table 3. Table A shows the comparative operating data for the Detroit residence for the past 2 summer seasons, and the economy obtained by taking full advantage of night air cooling is quite evident. As the proportion of the season's ice consumption used each month bore a close relationship to the percentage of degree hours above 85 F occurring in that month in the 1932 season, this comparison has been extended to include last summer, and is shown in Table B.

The author's conclusion that the most effective method of circulating air from outdoors at night is to open both the first and second story windows seems inconsistent with the curves shown in Fig. 7, and the air velocity data shown in Table 2. A comparison of curves Nos. 3 and 5, as shown in Fig. 7, indicates a more marked improvement due to the attic fan when pulling air through the second floor only, than when pulling air through windows in both stories. As night air cooling increases in effectiveness as the hour grows later, and as the occupants of the house spend most of the night in the upstairs sleeping rooms, it would seem best to confine the effect to the second floor where the occupants can receive its full benefit. Experience in Detroit residences indicates that the second floor bedrooms cannot be cooled very effectively by attic fans if the air flow is so divided between the 2 floors that the rate of air change is reduced to about one half of what it would be when pulling air through the second floor only. There is an advantage, however, in keep-

ing both the first and second story windows open until the hour the family retires. After that time, because of practical considerations, it seems unwise to keep the first floor windows open. It is not apparent, from the author's discussion of Fig. 7, whether the temperature drops shown for the attic fan used with a full window opening on both floors, as shown in Curve 4, are for the first or second story. It would be interesting to know whether or not these are second story temperatures.

TABLE A—COMPARATIVE OPERATING DATA FOR COOLING SEASONS OF 1932 AND 1933

	1932	1933
Number of days when fan or artificial cooling was used	22	32
Number of days when artificial cooling only was used	22	22
Number of hours when fan or artificial cooling was used	184	254
Number of hours when artificial cooling was used	135	134
Number of hours when fan only was used	49	120
Total equivalent amount of ice used, pounds	18,166	12,290
Average equivalent ice consumption per day of artificial cooling	826	559
Average equivalent ice consumption per hour of artificial cooling	134	91.5
Total number of degree hours above 85 F during summer	529	1408
Equivalent ice consumption per degree hour, pounds	34.3	8.7
Coil surface utilized for cooling, square feet	328	328
Power Used		
Electricity used for pump and fan, entire season, kwhr	94	142
Cost of Cooling for Season		
Ice (including equivalent of city water) at $5.00 per ton	$45.40	$30.60
Electricity at 2¼ cents per kwhr	2.12	3.20
Total operating cost	47.52	33.80
Average cost per day of artificial cooling	2.16	1.54
Average cost per hour of artificial cooling	0.352	0.252

TABLE B—RELATION BETWEEN COOLING REQUIREMENTS AND DEGREE HOURS

Month	Degree Hours Above 85 F	Ice Consumption Pounds	Per cent of Total Degree Hours	Per cent of Total Ice Consumption
1932				
July	339	8950	67	65
August	164	4782	33	35
Total	503	13,732[a]	100	100
1933				
June	716	6990	51	57
July	539	4000	38	33
August	39	800	3	6
September	110	500	8	4
Total	1404	12,290[b]	100	100

[a] Ice equivalent of city water used in June is *not* included.
[b] Ice equivalent of city water used during summer *is* included.

J. H. WALKER: I wish to discuss this paper from the economic standpoint. The authors used a method of cooling the research residence which was agreed to be a rather expensive method from the standpoint of operating cost, although the first cost of an ice system has been thought to be less than that of a mechanical system.

In Table C I have shown the figures from the authors' paper for the summer of 1932 giving the amount of ice used in tons for the straight ice system without any supplemental method, and for the basement fan and the attic fan supplementing the ice cooling. Using a price of $5.00 a ton, which is a fairly low price for ice, we get

the figures in the second column of Table C. Calculating the cost of power to operate the attic fan, the attic and basement fan, we arrive at the figures for the total cost shown in the fourth column. With electricity at 2¼¢ a kilowatt, which is perhaps a fair price, we get the figures shown in the last column of the table for the cost of operation of the mechanical system. The cost of operation when the ice system is supplemented by the basement or attic fan is reduced from $215 to $77 or $55 respectively. The cost of operating a mechanical system without supplementary fans is only $43. It would therefore seem most practicable to install the mechanical compressor without the complication and cost of the supplementary fans. The proper application of the attic fan is not to supplement artificial cooling, but to serve as a fairly effective method, by itself, for night-time cooling.

Table C—Comparison of Operating Costs of Various Cooling Methods—1932

	Tons of Ice Used	Cost of Ice at $5.00 Per Ton	Cost of Power for Fans	Total Cost	Total Cost with Mechanical System
No Supplementary Method	43	$215	..	$215	$43
Cooling supplemented by basement fan	14	$ 70	$7	$ 77	$21
Cooling supplemented by attic fan	10	$ 50	$5	$ 55	$15

F. J. Moelter: I wish to take exception to the figures presented in Table C, which do not show an accurate comparison for the reason that all costs such as depreciation, interest on investment, etc., were not taken into account and the power cost used was not representative of the country, thereby giving the impression that the cost of ice for the installation far exceeded the probable power cost, which is not the case in my opinion.

J. H. Walker: This entire discussion has dealt only with operating costs. Obviously, depreciation and interest on the investment must be considered in a summation of over-all costs. Our experience indicates, however, that the difference in first cost between an ice tank system and a mechanical compression system is not as great as one might assume.

The figures of $5.00 per ton for ice and 2¼¢ per kwhr for electricity are the actual prevailing prices for the experimental residence near Detroit. The figure of 2¼¢ per kwhr is not assumed to be the average cost of electricity throughout the country. Neither is the figure of $5.00 per ton for ice necessarily an average price. In some cases the price has been found to be higher.

If the use of ice for summer cooling is to become the important factor that its advantages merit, more attention must be given by the ice interests to quantity prices and convenient deliveries.

John Everetts: When using the temperature of 81 deg set up on our comfort chart in order to be comfortable, we should maintain approximately a 67.5 or 68 deg wet-bulb temperature. I should like to ask the author what the variation in wet-bulb temperature or deviation from the comfort optimum condition was in this particular fan cooling system or the attic fan cooling system supplemented with ice cooling.

S. Konzo: Mr. Walker has made the statement that with mechanical refrigeration it would be possible to make a reduction in the cost of summer cooling. That phase of summer cooling has, as yet, been untouched at the Research Residence and it is hoped that tests with a mechanical refrigeration plant can be included in the work for this coming summer.

In regard to the question as to why an 81 deg dry-bulb temperature was regarded as the starting condition, it was found that whenever the air temperature exceeded

81 deg, with accompanying relative humidities of approximately 50 to 60 per cent, the conditions could not be regarded as comfortable. In fact, it may be noted on the comfort chart that with an 81 deg dry-bulb temperature and with a relative humidity of from 50 to 60 per cent, the effective temperature is approximately 85 deg, which is the practical upper limit of the summer comfort zone.

In the work at the Residence it was found that as long as the air conditions could be maintained below the upper limit of the summer comfort zone, the occupants were comfortable. In cooler climates, where the body may not become acclimated to as high temperatures and humidities, the practical effective temperature may be much lower than 75. In the Residence work, therefore, 81 deg dry-bulb was regarded as the starting temperature for the ice cooling plant.

MR. EVERETTS: How were you able to obtain it?

MR. KONZO: Up to the time that the house temperature rose to 81 deg dry-bulb, no artificial cooling was used. When it tended to exceed that temperature, then the ice cooling plant was started and the air temperatures were prevented from exceeding that value.

MR. EVERETTS: Then the fact that Urbana has a wet-bulb of 72 or 73 deg, which would run a bit lower than New York, would mean you would not be able to do it in New York?

MR. KONZO: I think you have a slight misunderstanding as to just what we did. When the indoor temperature went up to 81 deg the wet-bulb temperature was allowed to go wherever it would. The wet-bulb ordinarily attained a value that was represented by 50 or 60 per cent humidity. When the ice cooling plant was started, then the dry-bulb temperature was not allowed to go up but was usually maintained constant at approximately 81 deg and the humidity was decreased to approximately 40 per cent after a few hours of operation.

MR. EVERETTS: Did it show any difference in cost for the entire season with attic fan ventilation supplemented with ice cooling than it did using the ice cooling alone without attic ventilation?

MR. KONZO: The relative ice consumptions for the two cases are presented in the paper and the estimate is made that instead of an ice requirement of 40 tons, the use of night cooling would reduce that requirement to 10 tons ice consumption.

MR. EVERETTS: The cost in dollars and cents?

MR. KONZO: The cost comparisons for the two cases would show a decided advantage for the case of night cooling supplemented with ice cooling, over straight ice cooling alone. Mr. Walker's discussion of comparative costs shows that to be the case. The operation of a fan during the night at a cost of about 25 cents may mean a saving of a half ton or more of ice costing 10 times as much as the electrical current for the fan.

No. 980

THERMODYNAMIC PROPERTIES OF MOIST AIR

By JOHN A. GOFF ‡ (NON-MEMBER), URBANA, ILL.

THERMODYNAMICS finds an important application in the field of heating and ventilating. Here, as in the field of power generation, it investigates and defines the properties of the media employed and describes the various ways in which desired changes of state can be accomplished with particular reference to the energy quantities involved. In its application to heating and ventilating, thermodynamics has to deal with a medium, namely, atmospheric air, which is somewhat more complex than the media usually employed in power generation due to the fact that it is one of variable composition. It may be expected, therefore, that atmospheric air will be found to possess thermodynamic properties not found in the simple media of power generation.

The heating and ventilating engineer already possesses sufficient information, mostly in the form of psychrometric charts, to enable him to deal effectively with most problems in his field involving properties of atmospheric air. This being true, it may appear quite unnecessary to attempt a further contribution to the subject. Still, much of this information, although capable of yielding approximate results of practical value, is, in some cases, strictly speaking, incorrect and sometimes leads to impossible conclusions if used without discretion. Consequently, an attempt to work out a thermodynamically consistent treatment of the properties of atmospheric air based on best available thermal data, written in the language of thermodynamics, but yet reduced to a form which admits of quick and easy application, may possibly find favor with even the more practical-minded engineer. The present paper is such an attempt.

DRY AIR

We begin with an important simplification. Thus, it will be recognized from the start that atmospheric air can be regarded as a mixture of only two constituents. The relative proportions of oxygen, nitrogen, carbon dioxide, etc., are either so small or subject to such small variations that they need not be considered separately. Instead, they may be treated collectively as com-

---

‡ Associate Professor of Thermodynamics, University of Illinois.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by A. P. Kratz.

ponents of a gas mixture of constant composition called, Dry Air. Dry air is one constituent of atmospheric air, and water is the other.

(1) *Composition.* For the sake of definiteness, the volume composition of dry air given in *International Critical Tables* (I, 393) may be used as standard. It is

$$\begin{aligned} N_2 &= 0.7803 \\ O_2 &= 0.2099 \\ A &= 0.0094 \\ CO_2 &= 0.0003 \\ H_2 &= 0.0001 \\ \hline & \ 1.0000 \end{aligned}$$

This volume composition is also a *mol* composition. Consequently, if each figure be multiplied by the molecular weight of the corresponding constituent and the resulting products added, their sum may be used as the molecular weight of dry air; thus, $m_a = 28.967$ (lb/mol).

(2) *Specific Volume.* Under the conditions of pressure and temperature ordinarily encountered in heating and ventilating, dry air may be assumed to behave like a perfect gas so that its specific volume v_a (ft³/lb), may be related to its pressure, p_a (lb/ft²), and absolute temperature, T (F), by the well-known equation

$$v_a = \frac{B_a \times T}{p_a} \tag{1}$$

in which $B_a = 1544/28.967 = 53.34$ (ft/F). Column 8, Table 1, gives values of v_a for a pressure, $p_a = 30$ in. Hg.

(3) *Specific enthalpy.*[1] Within the same range of temperature and pressure for which Eq. (1) is valid, the specific heat-at-constant pressure of dry air is a function of its temperature only. Accordingly, its specific enthalpy can be calculated from

$$h_a = \int (c_p)_a \, dT + \text{const} \tag{2}$$

in which the constant of integration can be chosen quite arbitrarily. For rough computations, $(c_p)_a$ is usually assigned the average value, 0.24 (Btu/lb F); but, for more accurate work, its actual variation with temperature should be taken into account. Column 5, Table I, gives values of h_a derived from specific heat data given in *International Critical Tables* (V, 81), the constant of integration having been adjusted to make all tabular values positive.

WATER: THERMODYNAMIC PROPERTIES

Although the moisture in atmospheric air is usually in the state of superheated vapor, sufficient cooling will always reduce it to a state of saturation. Consider, first of all, the thermodynamic properties of the saturated vapor.

Saturated Vapor

(1) *Vapor Pressure.* The pressure, p''_w of saturated water vapor is a function of the temperature only. The best experimental data on the relation between p''_w and t are probably those in *International Critical Tables* (III, 210-211). The actual vapor pressure is slightly affected by the presence of an indifferent gas such as dry air. Consequently, a correction easily estimated from formulae in *International Critical Tables* should be applied for the sake of accuracy.

[1] Enthalpy is defined by the equation $h = u + Apv$ in which u denotes intrinsic energy. Applying the energy equation of thermodynamics to a change of state *at constant pressure,* it turns out that the change of enthalpy gives exactly the quantity of heat absorbed. Enthalpy is sometimes called *heat content;* but this latter expression deserves the earliest possible abandonment for the fundamental reason that heat is an external performance, not a property, and as such, cannot be contained.

TABLE 1

Temperature (F)	Quality (P_0 = 30 in. Hg)		Specific Enthalpy of Liquid (or Solid) Water (Btu/pound)	Enthalpy (Btu/pound Dry Air) (P_0 = 30 in. Hg)			Volume (Ft.³/pound Dry Air) (P_0 = 30 in. Hg)			Wet-Bulb Depression (F) (P_0 = 30 in. Hg)		
	Weight of Saturated Vapor per Pound Dry Air	Correction Factor for Total Pressure *		Dry Air ($\mu = 0$)		Just-saturated Mixture ($\mu = 1$)	Dry Air ($\mu = 0$)		Just-saturated Mixture ($\mu = 1$)	Dry Air ($\mu = 0$)		Constant**
t	x''_0	a	h'_w	h'_0	$h''_0 - h'_0$	h''_0	v'_0	$v''_0 - v'_0$	v''_0	δ'	μc†	b
0	0.000783	0.0334	−158.8	20.39	0.83	21.22	11.56	0.01	11.57	3.35	....	0.016
2	0.000870	0.0334	−157.9	20.87	0.92	21.79	11.61	0.01	11.62	3.65	....	0.019
4	0.000965	0.0334	−157.0	21.35	1.03	22.38	11.66	0.02	11.68	3.98	....	0.022
6	0.001069	0.0334	−156.0	21.83	1.14	22.97	11.71	0.02	11.73	4.39	....	0.026
8	0.001183	0.0334	−155.1	22.31	1.26	23.57	11.76	0.02	11.78	4.72	....	0.030
10	0.001308	0.0334	−154.1	22.79	1.40	24.19	11.81	0.02	11.83	5.12	....	0.033
12	0.001446	0.0334	−153.2	23.27	1.55	24.82	11.86	0.03	11.89	5.55	....	0.036
14	0.001597	0.0334	−152.2	23.75	1.71	25.46	11.91	0.03	11.94	6.00	....	0.040
16	0.001762	0.0334	−151.3	24.23	1.88	26.11	11.96	0.03	11.99	6.48	....	0.044
18	0.001943	0.0334	−150.3	24.71	2.08	26.79	12.01	0.04	12.05	6.99	....	0.049
20	0.002140	0.0335	−149.3	25.19	2.29	27.48	12.06	0.04	12.10	7.52	....	0.054
22	0.002356	0.0335	−148.4	25.67	2.52	28.19	12.11	0.05	12.16	8.08	....	0.061
24	0.002591	0.0335	−147.4	26.15	2.78	28.93	12.16	0.05	12.21	8.66	....	0.069
26	0.002848	0.0335	−146.4	26.63	3.06	29.69	12.21	0.06	12.27	9.27	....	0.078
28	0.003128	0.0335	−145.4	27.11	3.36	30.47	12.26	0.06	12.32	9.91	....	0.087
30	0.003432	0.0335	−144.5	27.59	3.69	31.28	12.31	0.07	12.38	10.57	....	0.098
32	0.003763	0.0335	−143.5	28.07	4.05	32.12	12.36	0.08	12.44	11.26	....	0.109
33	0.003920	0.0335	1.0	28.31	4.22	32.53	12.39	0.07	12.46	(11.03)†	0.910	0.079
34	0.004082	0.0335	2.0	28.55	4.39	32.94	12.41	0.08	12.49	(11.36)	0.826	0.084
35	0.004251	0.0336	3.0	28.79	4.57	33.36	12.44	0.08	12.52	(11.70)	0.746	0.089
36	0.004425	0.0336	4.0	29.03	4.76	33.79	12.46	0.09	12.55	(12.04)	0.672	0.094
37	0.004606	0.0336	5.0	29.27	4.96	34.23	12.49	0.09	12.58	(12.39)	0.603	0.099
38	0.004792	0.0336	6.0	29.51	5.16	34.67	12.51	0.10	12.61	(12.74)	0.539	0.104
39	0.004985	0.0336	7.0	29.75	5.37	35.12	12.54	0.10	12.64	(13.10)	0.478	0.109

TABLE 1—*Continued*

40	0.005183	0.0336	8.1	29.99	5.59	35.58	12.56	0.11	12.67	(13.46)	0.421	0.115
41	0.005390	0.0336	9.1	30.23	5.81	36.04	12.59	0.11	12.70	(13.83)	0.369	0.120
42	0.005604	0.0336	10.1	30.47	6.05	36.52	12.61	0.12	12.73	(14.20)	0.320	0.126
43	0.005826	0.0336	11.1	30.71	6.29	37.00	12.64	0.12	12.76	(14.58)	0.274	0.131
44	0.006056	0.0336	12.1	30.95	6.54	37.49	12.66	0.13	12.79	(14.96)	0.231	0.137
45	0.006294	0.0337	13.1	31.19	6.80	37.99	12.69	0.13	12.82	(15.34)	0.191	0.142
46	0.006540	0.0337	14.1	31.43	7.07	38.50	12.71	0.14	12.85	(15.73)	0.154	0.148
47	0.006794	0.0337	15.1	31.67	7.35	39.02	12.74	0.14	12.88	(16.12)	0.119	0.153
48	0.007058	0.0337	16.1	31.91	7.63	39.54	12.76	0.15	12.91	(16.51)	0.087	0.159
49	0.007330	0.0337	17.1	32.15	7.93	40.08	12.79	0.15	12.94	(16.90)	0.057	0.164
50	0.007612	0.0337	18.1	32.39	8.24	40.63	12.81	0.16	12.97	(17.32)	0.029	0.172
51	0.007903	0.0337	19.1	32.63	8.56	41.19	12.84	0.16	13.00	(17.74)	0.003	0.178
52	0.008204	0.0337	20.1	32.87	8.89	41.76	12.86	0.17	13.03	18.16	....	0.185
53	0.008517	0.0337	21.1	33.11	9.23	42.34	12.89	0.18	13.07	18.58	....	0.191
54	0.008841	0.0337	22.1	33.35	9.59	42.94	12.91	0.19	13.10	19.01	....	0.198
55	0.009175	0.0338	23.1	33.59	9.95	43.54	12.94	0.19	13.13	19.44	....	0.205
56	0.009519	0.0338	24.1	33.83	10.33	44.16	12.96	0.20	13.16	19.88	....	0.212
57	0.009875	0.0339	25.1	34.07	10.72	44.79	12.99	0.20	13.19	20.32	....	0.219
58	0.01024	0.0339	26.1	34.31	11.12	45.43	13.01	0.22	13.23	20.77	....	0.226
59	0.01062	0.0339	27.1	34.55	11.54	46.09	13.04	0.22	13.26	21.22	....	0.233
60	0.01102	0.0339	28.1	34.79	11.97	46.76	13.06	0.24	13.30	21.67	....	0.241
61	0.01142	0.0340	29.1	35.03	12.42	47.45	13.09	0.24	13.33	22.13	....	0.248
62	0.01184	0.0340	30.1	35.27	12.88	48.15	13.11	0.25	13.36	22.60	....	0.256
63	0.01227	0.0340	31.1	35.51	13.35	48.86	13.14	0.26	13.40	23.07	....	0.264
64	0.01272	0.0340	32.1	35.75	13.84	49.59	13.17	0.26	13.43	23.54	....	0.272
65	0.01318	0.0340	33.1	35.99	14.35	50.34	13.19	0.28	13.47	24.02	....	0.280
66	0.01366	0.0341	34.1	36.23	14.88	51.11	13.22	0.29	13.51	24.51	....	0.289
67	0.01415	0.0341	35.1	36.47	15.42	51.89	13.24	0.30	13.54	24.99	....	0.298
68	0.01466	0.0341	36.1	36.71	15.98	52.69	13.27	0.31	13.58	25.49	....	0.307
69	0.01519	0.0342	37.1	36.94	16.57	53.51	13.29	0.33	13.62	25.98	....	0.316
70	0.01573	0.0342	38.1	37.18	17.17	54.35	13.32	0.33	13.65	26.48	....	0.326
71	0.01629	0.0342	39.1	37.42	17.79	55.21	13.34	0.35	13.69	26.99	....	0.335
72	0.01687	0.0342	40.1	37.66	18.42	56.08	13.37	0.36	13.73	27.50	....	0.345
73	0.01746	0.0343	41.1	37.90	19.08	56.98	13.39	0.38	13.77	28.01	....	0.355
74	0.01808	0.0343	42.1	38.14	19.76	57.90	13.42	0.39	13.81	28.53	....	0.365

TABLE 1—*Continued*

Temperature (F)	Quality (P_0=30 in. Hg) Weight of Saturated Vapor per Pound Dry Air	Quality (P_0=30 in. Hg) Correction Factor for Total Pressure *	Specific Enthalpy of Liquid (or Solid) Water (Btu/pound)	Enthalpy (Btu/pound Dry Air) (P_0=30 in. Hg) Dry Air (μ=0)	Enthalpy (Btu/pound Dry Air) (P_0=30 in. Hg)	Enthalpy (Btu/pound Dry Air) (P_0=30 in. Hg) Just-saturated Mixture (μ=1)	Volume (Ft.³/pound Dry Air) (P_0=30 in. Hg) Dry Air (μ=0)	Volume (Ft.³/pound Dry Air) (P_0=30 in. Hg)	Volume (Ft.³/pound Dry Air) (P_0=30 in. Hg) Just-saturated Mixture (μ=1)	Wet-Bulb Depression (F) (P_0=30 in. Hg) Dry Air (μ=0)	Wet-Bulb Depression (F) (P_0=30 in. Hg)	Wet-Bulb Depression (F) (P_0=30 in. Hg) Constant**
t	x''_0	a	h'_w	h'_0	h''_0-h'_0	h''_0	v'_0	v''_0-v'_0	v''_0	δ'	μc†	b
75	0.01871	0.0343	43.1	38.38	20.45	58.83	13.44	0.41	13.85	29.05	....	0.375
76	0.01936	0.0344	44.1	38.62	21.18	59.80	13.47	0.42	13.89	29.57	....	0.385
77	0.02004	0.0344	45.1	38.86	21.92	60.78	13.49	0.44	13.93	30.10	....	0.395
78	0.02073	0.0345	46.1	39.10	22.69	61.79	13.52	0.45	13.97	30.63	....	0.406
79	0.02145	0.0345	47.1	39.34	23.49	62.83	13.54	0.47	14.01	31.17	....	0.417
80	0.02219	0.0345	48.1	39.58	24.31	63.89	13.57	0.48	14.05	31.71	....	0.428
81	0.02295	0.0346	49.1	39.82	25.15	64.97	13.59	0.50	14.09	32.26	....	0.438
82	0.02373	0.0346	50.1	40.06	26.02	66.08	13.62	0.52	14.14	32.81	....	0.449
83	0.02454	0.0347	51.1	40.30	26.92	67.22	13.64	0.54	14.18	33.36	....	0.460
84	0.02538	0.0347	52.0	40.54	27.84	68.38	13.67	0.56	14.23	33.91	....	0.471
85	0.02624	0.0347	53.0	40.78	28.80	69.58	13.69	0.58	14.27	34.47	....	0.483
86	0.02713	0.0348	54.0	41.02	29.79	70.81	13.72	0.60	14.32	35.04	....	0.495
87	0.02805	0.0348	55.0	41.26	30.81	72.07	13.74	0.62	14.36	35.60	....	0.508
88	0.02899	0.0349	56.0	41.50	31.87	73.37	13.77	0.64	14.41	36.17	....	0.521
89	0.02997	0.0349	57.0	41.74	32.96	74.70	13.79	0.67	14.46	36.75	....	0.534
90	0.03097	0.0350	58.0	41.98	34.08	76.06	13.82	0.69	14.51	37.33	....	0.548
91	0.03201	0.0351	59.0	42.22	35.24	77.46	13.84	0.72	14.56	37.91	....	0.562
92	0.03308	0.0351	60.0	42.46	36.44	78.90	13.87	0.74	14.61	38.50	....	0.576
93	0.03419	0.0352	61.0	42.70	37.67	80.37	13.89	0.77	14.66	39.09	....	0.590
94	0.03533	0.0352	62.0	42.94	38.94	81.88	13.92	0.79	14.71	39.68	....	0.604
95	0.03651	0.0353	63.0	43.18	40.25	83.43	13.94	0.82	14.76	40.27	....	0.618
96	0.03772	0.0354	64.0	43.42	41.60	85.02	13.97	0.85	14.82	40.86	....	0.632
97	0.03897	0.0354	65.0	43.66	42.99	86.65	14.00	0.87	14.87	41.46	....	0.646
98	0.04025	0.0355	66.0	43.90	44.42	88.32	14.02	0.91	14.93	42.06	....	0.661
99	0.04157	0.0356	67.0	44.14	45.90	90.04	14.05	0.93	14.98	42.66	....	0.675

TABLE 1—*Continued*

100	0.04293	0.0356	68.0	44.38	47.41	91.79	14.07	0.97	15.04	43.27	....	0.690
101	0.04434	0.0357	69.0	44.62	48.97	93.59	14.10	1.00	15.10	43.88	....	0.705
102	0.04578	0.0358	70.0	44.86	50.59	95.45	14.12	1.04	15.16	44.49	....	0.720
103	0.04727	0.0359	71.0	45.10	52.26	97.36	14.15	1.07	15.22	45.11	....	0.735
104	0.04881	0.0359	72.0	45.34	53.98	99.32	14.17	1.11	15.28	45.73	....	0.750
105	0.05040	0.0360	73.0	45.58	55.76	101.34	14.20	1.15	15.35	46.35	....	0.765
106	0.05204	0.0361	74.0	45.81	57.60	103.41	14.22	1.19	15.41	46.98	....	0.781
107	0.05373	0.0362	75.0	46.05	59.49	105.54	14.25	1.23	15.48	47.61	....	0.797
108	0.05547	0.0363	76.0	46.29	61.44	107.73	14.27	1.27	15.54	48.24	....	0.813
109	0.05726	0.0364	77.0	46.53	63.45	109.98	14.30	1.31	15.61	48.88	....	0.830
110	0.05910	0.0365	78.0	46.77	65.53	112.30	14.32	1.36	15.68	49.51	....	0.847
111	0.06100	0.0366	79.0	47.01	67.67	114.68	14.35	1.40	15.75	50.15	....	0.864
112	0.06296	0.0367	80.0	47.25	69.87	117.12	14.37	1.45	15.82	50.80	....	0.882
113	0.06498	0.0368	81.0	47.49	72.15	119.64	14.40	1.50	15.90	51.44	....	0.900
114	0.06707	0.0369	82.0	47.73	74.49	122.22	14.42	1.55	15.97	52.09	....	0.918
115	0.06922	0.0370	83.0	47.97	76.92	124.89	14.45	1.60	16.05	52.73	....	0.937
116	0.07143	0.0372	84.0	48.21	79.42	127.63	14.47	1.66	16.13	53.39	....	0.956
117	0.07372	0.0373	85.0	48.45	82.00	130.45	14.50	1.71	16.21	54.04	....	0.975
118	0.07609	0.0374	86.0	48.69	84.66	133.35	14.52	1.77	16.29	54.70	....	0.995
119	0.07854	0.0375	86.9	48.93	87.40	136.33	14.55	1.83	16.38	55.36	....	1.015
120	0.08106	0.0377	87.9	49.17	90.22	139.39	14.57	1.90	16.47	56.02	....	1.03
121	0.08365	0.0378	88.9	49.41	93.13	142.54	14.60	1.95	16.55	56.68	....	1.05
122	0.08630	0.0380	89.9	49.65	96.14	145.79	14.62	2.03	16.65	57.34	....	1.07
123	0.08903	0.0381	90.9	49.89	99.25	149.14	14.65	2.09	16.74	58.02	....	1.09
124	0.09185	0.0382	91.9	50.13	102.42	152.55	14.67	2.16	16.83	58.69	....	1.11
125	0.09478	0.0384	92.9	50.37	105.73	156.10	14.70	2.23	16.93	....	....	....
126	0.09779	0.0386	93.9	50.61	109.15	159.76	14.72	2.31	17.03	....	....	....
127	0.1009	0.0387	94.9	50.85	112.67	163.52	14.75	2.38	17.13	....	....	....
128	0.1042	0.0389	95.9	51.09	116.29	167.38	14.77	2.47	17.24	....	....	....
129	0.1075	0.0391	96.9	51.33	120.03	171.36	14.80	2.55	17.35	....	....	....

* For use in the formula, $z'' = \frac{z''_0}{1 + a\,(P - P_0)}$

** For use in the empirical equation, $\delta = \frac{1 - \mu}{1 + b\mu}\,\delta'$ or $\mu = \frac{\delta'/\delta - 1}{\delta'/\delta + b}$.

† The figure in parenthesis is not the true wet-bulb depression of dry air. It can be used in the empirical equation, however, provided the percentage humidity is greater than the critical value μ_c.

(2) *Specific Volume.* The specific volume of saturated water vapor can also be calculated from the perfect gas equation,

$$v''_w = \frac{B_w \times T}{p''_w} \tag{3}$$

as in the case of dry air with this difference, however, that for accurate work, the constant B_w requires some adjustment in order to reproduce the experimental data at the higher temperatures. For ordinary calculations, the theoretical value $B_w = 1544/18.016 = 85.70$ (ft/F) may be used throughout.

(3) *Specific Enthalpy.* Values for the specific enthalpy of saturated water vapor are to be found in the 1932 Knoblauch Tables (*Tabellen und Diagramme für Wasserdampf*). These tables are recommended because there is reason to think that they can be extrapolated to temperatures below 32 F with somewhat more certainty than others. This extrapolation is made by means of the equation,

$$h''_w = 1074.4 + 0.440\ (t - 32)\ \text{(Btu/lb)} \tag{4}$$

which appears to give correct results for temperatures below about 86 F.

Superheated Vapor

Direct experimental data on the thermodynamic properties of superheated water vapor are lacking for the range of temperature and pressure encountered in heating and ventilating. However, the assumption that the constant B_w and the specific enthalpy h_w are functions of the temperature [2] only can be relied upon to give sufficiently accurate results. This assumption implies, for example, that Eq. (4) also gives the specific enthalpy of the superheated water vapor (for temperatures below 86 F).

Liquid (or Solid)

Within the limits of accuracy desired, there is seldom any need of considering the specific volume of the liquid (or solid). The specific enthalpy, however, is an important item. It is denoted by h'_w and is listed in Column 4, Table 1.

MIXTURES OF DRY AIR AND WATER: CLASSIFICATION

Using the word *water* to refer to the substance H_2O without regard to the particular phase or phases in which it may appear, we denote by z the ratio *by weight* of water to dry air in a given mixture and call z the *quality* [3] of this mixture. It is convenient to study mixtures of dry air and water, in general, under the following classification:

(1) *Saturated Mixtures* in which the water appears in at least two phases; namely, saturated vapor and saturated liquid if the temperature of the mixture is above 32 F; or saturated vapor and saturated solid (ice) if the temperature of the mixture is below 32 F. Let z' denote the ratio *by weight* of liquid (or solid) to dry air and z'', the weight of saturated vapor per pound of dry air; then $z' + z'' = z$.

(2) *Just-saturated Mixtures* in which the amount of liquid (or solid) is too small to contribute to the thermal properties of the mixture otherwise than by merely insur-

[2] Strictly speaking, this assumption violates the laws of thermodynamics; for, if B_w varies with the temperature, h_w must vary with the volume. At higher temperatures, the error involved in this assumption may become significant (1 per cent or more), but at lower temperatures it is negligible.

[3] The term *humidity* is widely used to denote the ratio by weight of water vapor to dry air. What is desired here is a term to denote the ratio by weight of water to dry air even when a considerable portion of the water is in the liquid or solid state.

ing that the vapor is actually saturated. The quality of a just-saturated mixture is z''.

(3) *Unsaturated Mixtures* in which the water appears in only one phase, namely, superheated vapor. The quality of such a mixture is z. Let z'' be the quality of the corresponding just-saturated mixture having the same temperature t (and same total pressure P) and let z be compared with z''. The ratio z/z'' is called the *percentage humidity* of the given unsaturated mixture and is denoted by the letter μ.

DALTON'S LAW

According to Dalton's Law, the partial pressure of any constituent of a mixture of *gases* is exactly equal to the pressure that that constituent would exert if alone in the whole volume of the mixture and at the same temperature. It has been pointed out (Jakob, *Zeitschrift für Physik,* 41 [1927] 737) that Dalton's Law only applies strictly to mixtures of perfect gases; but its application to atmospheric air will not involve significant error.

Just-Saturated Mixtures

Consider, first of all, any just-saturated mixture, temperature t, total pressure P. The percentage humidity of such a mixture is unity, by definition. The partial pressures, p_a (dry air) and p''_w (saturated water vapor) satisfy the equation,

$$p_a + p''_w = P \tag{5}$$

Moreover, it is always assumed that both dry air and saturated water vapor occupy *independently* the whole volume of the mixture; hence,

$$v_a = z''\, v''_w = v'' \tag{6}$$

in which v'' denotes the volume of the given just-saturated mixture (per pound of dry air) and z'', its quality.

(1) *Quality.* To apply Dalton's Law, calculate v_a and v''_w from Eq. (1) and (3), respectively, and, by making use of (5), write Eq. (6) as follows,

$$\frac{B_a T}{(P - p''_w)} = z''\,\frac{B_w T}{p''_w}$$

whence,

$$z'' = \frac{B_a}{B_w}\,\frac{p''_w}{(P - p''_w)} \tag{7}$$

Equation (7) affords the means of determining the quality of a given just-saturated mixture. Column 2, Table 1, gives values of z'' for a total pressure $P_0 = 30$ in. Hg as calculated from (7) using values of p''_w corrected for the presence of dry air as explained before.

(2) *Volume.* Dalton's Law may be applied in a slightly different way to calculate the volume v'' (per pound of dry air) of a just-saturated mixture. Thus p_a and p''_w are calculated from Eq. (1) and (3) and introduced into (5) as follows,

$$\frac{B_a \times T}{v_a} + \frac{B_w \times T}{v''_w} = P$$

Then, making use of (6),

$$v'' = \frac{B_a \times T}{P} + z''\,\frac{B_w \times T}{P} \tag{8}$$

Equation (8) shows that v'' is made up of two parts; the first is the contribution of the dry air itself to the total volume of the mixture and may be denoted by v' (when it is desired to refer to the specific volume of the liquid—or solid—the sub-

script W is affixed; thus, v'_w); the second is the contribution of the saturated vapor. Columns 8, 9, and 10, Table 1, give values of v', $(v'' - v')$ and v'', respectively, all for a total pressure $P_0 = 30$ in. Hg.

(3) *Enthalpy.* The enthalpy of a mixture of perfect gases is exactly the sum of the enthalpies of its constituents. This statement, like Dalton's Law, is only approximately correct for mixtures of actual gases and vapors. Still, its application to atmospheric air involves no significant error, so,

$$h'' = h_a + z'' h''_w \tag{9}$$

for a just-saturated mixture. Here again, the contribution of the dry air itself is denoted by h'; that is, $h' = h_a$. Columns 5, 6, and 7, Table 1, give values of h', $(h'' - h')$ and h'', respectively, all for a total pressure, $P_0 = 30$ in. Hg.

Saturated Mixtures

In a saturated mixture, the liquid (or solid) occupies exclusively a portion of the total volume. This portion is usually so small, however, that it may be neglected. The contribution of the liquid (or solid) to the total enthalpy of the mixture cannot be neglected, however. Let z' denote the weight of this liquid (or solid) per pound of dry air; then, its contribution to the enthalpy of the mixture per pound of dry air is $z'h'_w$ and we have,

$$h = h'' + z'h'_w \tag{10}$$

As an example showing the use that can be made of Table 1 in this connection, let it be required to find the volume and enthalpy of a saturated mixture, temperature 70 F, total pressure 30 in. Hg, quality 0.02. From Table 1, $z'' = 0.01573$ whence $z' = 0.00427$; therefore, the enthalpy is

$$h = 54.35 + 0.00427 \times 38.1 = 54.51 \text{ Btu/lb dry air.}$$

The volume of the mixture is obtained directly from the table, if no allowance for the volume occupied by the liquid itself is made. Thus

$$v = 13.65 \text{ ft}^3\text{/lb dry air.}$$

Unsaturated Mixtures

Remembering that the quality of an unsaturated mixture may be expressed in terms of its percentage humidity by the simple equation, $z = \mu z''$, in which z'' refers to the quality of the corresponding just-saturated mixture having the *same* temperature t and same total pressure P, simple expressions for volume and enthalpy in terms of percentage humidity can be indicated as follows:

(1) *Volume.* The volume (per pound of dry air) is

$$v = v' + \mu(v'' - v') \tag{11}$$

(2) *Enthalpy.* The enthalpy (per pound of dry air) is

$$h = h' + \mu\,(h'' - h') \tag{12}$$

The arrangement of Table 1 is designed to make full use of the simple linear relations expressed by Eq. (11) and (12). For example, let it be required to find the volume and enthalpy of an unsaturated mixture, temperature 70 F, percentage humidity 40 per cent, total pressure 30 in. Hg. Then

$$v = 13.32 + 0.4 \times 0.33 = 13.35 \text{ ft}^3\text{/lb dry air.}$$

$$h = 37.18 + 0.4 \times 17.17 = 44.05 \text{ Btu/lb dry air.}$$

(3) *Dew-point.* Any unsaturated mixture possesses two additional properties of practical importance; namely, dew-point and wet-bulb temperatures. Let the given mixture be cooled at constant total pressure P and constant quality z until its contained moisture just starts to condense. The temperature reached is called the *dew-*

point of the given unsaturated mixture. Dew-points are easily obtained by interpolation from Table 1. Thus, for a mixture, temperature 70 F, percentage humidity 40 per cent, total pressure 30 in. Hg, the quality is

$$z = 0.4 \times 0.01573 = 0.006292.$$

But, 0.006292 is the quality of a just-saturated mixture at 45.0 F. Hence, 45.0 F is the dew-point, t_d, of the given mixture.

(4) *Wet-bulb.* In a later paragraph, a thermodynamic definition of wet-bulb temperature will be given. For the present, however, the use that can be made of Table 1 in computing wet-bulb temperatures on the basis of an empirical equation whose constants have been adjusted to give accurate results, will be indicated. Thus, Column 11 gives the *wet-bulb depression* δ' of dry air and Column 12 gives the value of the constant b both appearing in the empirical equation,

$$\delta = \frac{1 - \mu}{1 + b\mu}\delta' \qquad (13)$$

in which δ is the wet-bulb depression of the given mixture, percentage humidity μ, total pressure 30 in. Hg.

For atmosphere air, temperature 70 F, percentage humidity 40 per cent, total pressure 30 in. Hg, the wet-bulb depression is

$$\delta = \frac{0.6 \times 26.48}{1.1304} = 14.05$$

Hence, the wet-bulb temperature itself is

$$t_w = 55.95 \text{ F}.$$

Between 32 F and 51.13 F, Eq. (13) can only be used for values of percentage humidity greater than a certain critical value μ_c at which ice would form in the wet-bulb thermometer. Even when this restriction is observed, it is necessary to use a fictitious value for δ' in Eq. (13), the fact that this value is not the true wet-bulb depression of dry air being indicated by parentheses in Table 1. For example, the wet-bulb temperature of a mixture, temperature 40 F, percentage humidity 60 per cent, is

$$t_w = 40 - \frac{0.4 \times 13.46}{1.069} = 34.97 \text{ F}.$$

The wet-bulb of a mixture at the same temperature but percentage humidity less than the critical value, 0.421, cannot be calculated from Table 1 directly.

CORRECTIONS FOR TOTAL PRESSURE

All items in Table 1 are computed for a total pressure, $P_0 = 30$ in. Hg. Frequently, however, it is desired to obtain values for a different total pressure, say $P = P_0 + \Delta P$. This is easily accomplished through the use of the constant a, Column 3. Thus, for a mixture, percentage humidity μ, total pressure P;

$$\text{(a)} \quad z = \frac{\mu z_0''}{1 + a\,(P - P_0)}$$

$$\text{(b)} \quad h = h_0' + \frac{\mu(h_0'' - h_0')}{1 + a\,(P - P_0)}$$

$$\text{(c)} \quad v = \frac{P_0}{P}\left[v_0' + \frac{\mu(v_0'' - v_0')}{1 + a\,(P - P_0)}\right]$$

The corresponding total-pressure correction for the dew-point and wet-bulb have not yet been worked out. In this respect, Table 1, is still somewhat incomplete.

As an example, let it be required to find the quality, enthalpy, and volume of a mixture, temperature 70 F, percentage humidity 40 per cent, total pressure 25 in. Hg. They are:

(a) $z = \dfrac{0.4 \times 0.01573}{0.829} = 0.007589$ lb/lb dry air

(b) $h = 37.18 + \dfrac{0.4 \times 17.17}{0.829} = 45.47$ Btu/lb dry air

(c) $v = \dfrac{30}{25}\left[13.32 + \dfrac{0.4 \times 0.33}{0.829}\right] = 16.18$ ft^3/lb dry air

Mixing Processes

Having investigated and defined the various thermodynamic properties of mixtures of dry air and water of practical importance, an analysis of the various processes involved in problems of heating and ventilating can be made. Consider a continuous-flow mixing process in which $m_1(1+z_1)$ lb per minute of Mixture 1, $m_2(1+z_2)$ lb per minute of Mixture 2, and m_w lb per minute of pure water (solid, liquor or vapor) unite to form $(1+z)$ lb per minute of Resulting Mixture. Weight balances for the dry air and water separately require,

$$m_1 + m_2 = 1 \tag{14a}$$

$$m_1z_1 + m_2z_2 + m_w = z \tag{14b}$$

If all velocities are low so that no allowance for kinetic energies need be made, then the total enthalpy of the Resulting Mixture will equal the sum of the enthalpies of the component mixtures plus any heat absorbed from outside. Let the amount of heat thus absorbed from outside *per pound of dry air in the Resulting Mixture* be q Btu; then,

$$m_1h_1 + m_2h_2 + m_wh_w = h - q \tag{14c}$$

in which h_1, h_2 and h are enthalpies per pound of dry air and h_w is the specific enthalpy of the water. Equations (14) may be said to describe the mixing process under consideration.

For some applications, Eq. (14) are used directly; for others, it is convenient to solve them for m_1, m_2 and m_w. This solution, for the special case of adiabatic mixing, $q = 0$, is

$$m_1 = \frac{(h-h_2) - h_w(z-z_2)}{(h_1-h_2) - h_w(z_1-z_2)} \tag{15a}$$

$$m_2 = 1 - m_1 \tag{15b}$$

$$m_w = (z-z_2) - m_1(z_1-z_2) \tag{15c}$$

Finally, two special applications of Eq. (15) are frequently used, namely:

Case I: $m_w = 0$

$$m_1 = \frac{h-h_2}{h_1-h_2} = \frac{z-z_2}{z_1-z_2} \tag{16}$$

$m_2 = 1 - m_1$

Case II: $m_2 = 0$

$$m_w = \frac{h-h_1}{h_w} = z - z_1 \tag{17}$$

$m_1 = 1$

The use of these equations may be illustrated by the following examples: (1) Outside Air, temperature 0 F, percentage humidity 25 per cent, is mixed adiabatically with recirculated Inside Air, temperature 70 F, percentage humidity 40 per cent. What volume of Inside Air will be required per cubic foot of Outside Air in order that the resulting temperature will be just 30 F?

Find the resulting percentage humidity. The total pressure may be assumed to be 30 in. Hg.

From Table 1, the following data are obtained:

	Outside Air	*Inside Air*	*Resulting Mixture*
Quality:	$z_1 = 0.000196$	$z_2 = 0.006292$	$z = 0.003432\mu$
Volume:	$v_1 = 11.56$	$v_2 = 13.45$	
Enthalpy:	$h_1 = 20.60$	$h_2 = 44.05$	$h = 27.59 + 3.69\mu$

Since the mixing is adiabatic and, in addition, $m_w = 0$, Eq. (16) may be used directly; thus

$$m_1 = \frac{16.46 - 3.69\mu}{23.45} = \frac{0.006292 - 0.003432\mu}{0.006096}$$

from which

$$\mu = 0.814 \qquad m_1 = 0.574 \qquad m_2 = 0.426$$

Finally, the volume of Inside Air per cubic foot of Outside Air is

$$\frac{0.426 \times 13.45}{0.574 \times 11.56} = 0.864 \text{ cu ft}$$

(2) Referring to Example 1, suppose Inside Air and Outside Air to be mixed adiabatically in the proportion of 1 cu ft of the former to 2 cu ft of the latter. Find the temperature and percentage humidity of the Resulting Mixture.

In the first place,

$$13.45\, m_2 = 5.78\, m_1$$

$$m_2 = 1 - m_1$$

so that

$$m_1 = 0.700$$

$$m_2 = 0.300$$

Eq. (14) may now be used directly (with $q = 0$) to determine z and h for the Resulting Mixture. They are

$$z = 0.700 \times 0.000196 + 0.300 \times 0.006292 = 0.002025$$

$$h = 0.700 \times 20.60 + 0.300 \times 44.05 = 27.63$$

At $t = 21.2$ F (found by trial and error)

$$h = 25.48 + \frac{0.002025}{0.002270} \times 2.43 = 27.64 \text{ (should be 27.63)}$$

Hence,

$$t = 21.2 \text{ F and } \mu = \frac{0.002025}{0.002270} = 0.892$$

WET-BULB TEMPERATURE: THERMODYNAMIC DEFINITION

Every unsaturated mixture exhibits an important thermodynamic property called *wet-bulb temperature*. Suppose that the given mixture is made to move

continuously over the surface of a thin film of liquid (or solid) water. This motion will cause continuous evaporation from the surface and the water thus evaporated will diffuse through a bounding layer of air, in which are established more or less steep temperature and quality gradients, to the given mixture. Moreover, these temperature and quality gradients in the bounding layer will reduce the given mixture to a just-saturated mixture right at the liquid (or solid) surface. The temperature of this just-saturated mixture may at first be quite different from that of the liquid (or solid) itself; but, if the film of liquid (or solid) is thin enough not to resist the necessary temperature change, this difference will gradually disappear and a steady state will ultimately be reached in which the temperature of the liquid (or solid) and that of the adjacent just-saturated mixture are sensibly equal. This steady temperature assumed by the liquid (or solid) is the wet-bulb temperature of the given unsaturated mixture. It may be denoted by t_w.

To reach a thermodynamic definition of wet-bulb temperature, it is customary to assume that the whole process of evaporation and subsequent mixing is adiabatic. This amounts to assuming that the effects of radiation can be excluded. To the evaporation itself, Eq. (17) applies as follows:

$$\frac{(h'')^* - h}{(z'')^* - z} = (h'_w)^*$$

in which the asterisk indicates that the designated quantities are to be evaluated at the wet-bulb temperature.

To the subsequent mixing, Eq. (16) applies as follows:

$$\frac{(h'')^* - h}{(z'')^* - z} = \frac{(h'')^* - h_1}{(z'')^* - z_1}$$

in which (h_1, z_1) is the state of the air whose wet-bulb is being measured. Combining,

$$\frac{(h'')^* - h_1}{[(z'')^* - z_1]\,(h'_w)^*} = 1 \qquad (18)$$

Equation (18) is the thermodynamic definition of the wet-bulb temperature used in evaluating the constants δ' and b appearing in Table 1. The derivation given here is that of Mollier (*Das ix-Diagramm für Dampfluftgemische:* Stodola Festschrift, 1929). It is based upon the so-called convection theory which has just recently received critical examination by Arnold (*Physics:* 4, 1933, 255) who contends that it fails to give correct results for liquids other than water.

APPLICATIONS TO AIR CONDITIONING

(1) *Heating and Humidifying—No Recirculation.* Air is to be supplied to a building at 70 F, percentage humidity 40 per cent. The outside temperature is 0 F, percentage humidity 25 per cent.

The dew-point of the Inside Air is 45.0 F. Consequently, if the Outside Air can be reduced to a just-saturated mixture at 45.0 F, subsequent addition of heat will bring it to 70 F and 40 per cent percentage humidity as required. In actual practice, this is accomplished by: (1) increasing the wet-bulb temperature of Outside Air to 45 F by the addition of heat from so-called tempering coils; (2) introducing water from a spray at 45 F to reduce the

Outside Air to a just-saturated mixture at 45 F—this is called *adiabatic saturation;* (3) adding heat from reheater coils to produce Inside Air. The following data may be assembled from Table 1:

	Outside Air	After Tempering Coils	After Sprays	Inside Air
Temperature F	0	(72.1)	45.0	70
Percentage Humidity	25%	....	100%	40%
Quality	0.000196	0.000196	0.006292	0.006292
Enthalpy	20.60	(37.91)	37.99	44.05
Volume	11.56	....	....	13.45

Equations (17) apply to process (1) plus (2) provided, of course, that h be replaced by $(h - q)$; thus,

$$m_w = \frac{(37.99 - q) - 20.60}{13.1} = 0.006292 - 0.000196$$

from which, $m_w = 0.006096$ and $q = 17.31$ Btu/lb dry air. The enthalpy of the mixture leaving the tempering coils is evidently 37.91 Btu/lb dry air $(= 20.60 + 17.31)$ and its quality is 0.000196; hence, its temperature is easily found to be 72.1 F.

The tempering coils must add 17.31 Btu of heat, the spray, 0.006096 lb of water at 45, and the reheater, 6.06 Btu of heat—all per pound of dry air. If 15,000 cfm of Inside Air is required, only $15{,}000 \dfrac{11.56}{13.45} = 12{,}900$ cfm of Outside Air need be handled.

(2) *Heating and Humidifying—With Recirculation.* Use the data of the preceding example, but allow sufficient recirculation to make the use of the tempering coils unnecessary. Equations (15) apply directly as follows:

$$m_1 = \frac{(37.99 - 20.60) - 13.1\ (0.006292 - 0.000196)}{(44.05 - 20.60) - 13.1\ (0.006292 - 0.000196)} = 0.741$$

$$m_2 = 0.259$$

$$m_w = 0.006096 - 0.741 \times 0.006096 = 0.00158$$

Therefore, if 15,000 cfm of Inside Air is required, $15{,}000 \times 0.741 = 11{,}100$ cfm must be recirculated with $15{,}000 \times 0.259 \times \dfrac{11.56}{13.45} = 3{,}340$ cfm of Outside Air. The heat required from the heater is the same as before; the amount of 45 deg water to be furnished from the spray is 0.00158 lb/lb dry air.

(3) *Cooling and Dehumidifying—No Recirculation.* Air is to be supplied to a building at 80 deg, percentage humidity 35 per cent. The outside temperature is 90 deg, percentage humidity 49 per cent.

The usual scheme for accomplishing the desired change is: (1) cool the Outside Air to the dew-point of the Inside Air by external refrigeration in the form of a cold water spray; (2) remove the liquid thus formed by passing over eliminator plates; (3) reheat the resulting just-saturated mixture to the

desired Inside Air conditions. The following data can be obtained from Table 1:

	Outside Air	After Spray	After Eliminator	Inside Air
Temperature F	90	50.5	50.5	80
Percentage Humidity	49%	....	100%	35%
Dew-point F	....	....	50.5	50.5
Quality	0.015175	0.015175	0.007767	0.007767
Enthalpy	58.69	41.07	40.93	48.09
Volume	14.16	....	....	13.74

Therefore, the refrigeration required is $58.69 - 41.07 = 17.63$ Btu/lb dry air, the heat added from the reheater is $48.09 - 40.93 = 7.16$ Btu/lb dry air, and the moisture removed by the eliminator is $0.015175 - 0.007767 = 0.007408$ lb/lb dry air.

DISCUSSION

A. A. Adler: I am pleased to note that Professor Goff has undertaken the task of clarifying some of our fundamental data as well as their expression in our future literature on air conditioning. I wish to point out some defects in our current presentation so that our more modern writers will see fit to correct ideas on some matters which are commonly accepted by practicing engineers as true beyond the shadow of a doubt. Before we can exercise discretion in the use of formulas we must know where they should be made to apply.

One set of phrases should promptly be dropped from our literature. These are Boyle's *law;* Gay-Lussac's *law; perfect gas* and *ideal gas.* These experimenters and their equipment, limited in range, and made at a time when experimental physics was without modern refinements, were probably justified in assuming that they had discovered natural laws. When these so-called *laws* are combined they result in the equation:

$$PV = RT \tag{1}$$

which is frequently referred to as the *perfect* or *ideal gas* equation. We know that this equation is not true at or near liquefaction and the trouble is (contrary to the usual belief) not with the gas but with the equation. The mathematics leading to Equation (1) is flawless, a characteristic of all mathematics. You cannot be partly right or partly wrong in mathematics. It is the only exact science. Such subjects as political economy, economics, psychology cannot by any stretch of the imagination be considered sciences. Physics and chemistry are usually considered exact sciences because accurate predictions are possible if all factors are considered. In this connection I would say that we should use such terms as: Let it be assumed that the relation between pressure, volume and temperature of a gas might be expressed with sufficient accuracy for the problem under consideration by the relation $PV = RT$. This would be doing the immature student a service because it is difficult to assume a *perfect gas* and then follow it with a warning that "departures from the law" always take place.

An improved gas equation was made by Van der Waals who wrote the equation: $(P + \frac{a}{V^2})(V - b) = RT$, where a/V^2 is to be interpreted as a minimum pressure which exists under any and all conditions due to the molecular attraction and b as

the molecular volume which is very small in all gases. Even this equation is based on Newtonian mechanics and fails when applied to atomic dimensions as is shown by the recent work done in electromechanics as applied to the breaking up of atoms.

The specific heat also should be offered only as a convenient artifice in determining the amount of heat involved in a process when a given weight of a substance is raised a given temperature range. I once noted a computation in a text book which showed an unbelievably high temperature reached as a result of burning pure carbon with pure oxygen and using the specific heat at ordinary temperatures as a universal constant.

While I have no specific comments to make relative to Dalton's Law I think it deserves a searching analysis in the light of the experimental evidence now available. Computations based on this as well as those on the gas equations should state clearly whether the figures arrived at are the result of computations by a stated formula or are the result of direct measurement. In all dubious cases the two should be put side by side so that discrepancies are not merely surmised but clearly noted.

In this connection I would like to offer a thought which may or may not be of importance. Relative to the three phases of matter, could it not be possible that they are not entirely distinct? For example, if an evacuated vessel has liquid introduced, experiment shows that at every temperature there is a corresponding vapor pressure. The question might be asked: "Does this vapor exist only in the space above the liquid or is it also diffused in the liquid?" A second question might be raised: "Is the liquid phase merely the solid phase (leaving only gaseous and solid phases) dispersed in the vapor phase under the conditions of dynamic equilibrium?" If these answers are in the positive, we might have a somewhat better concept of evaporation, saturated vapors and associated phenomena. I am not entirely convinced that this thought is correct but I offer it to Professor Goff and others interested for what it may be worth.

I hope that Professor Goff will take these comments in the friendly spirit in which they are offered. All our fundamental notions should be given a periodic housecleaning, for only in this way is progress made. It is not an engagement in personalities but merely a contest searching for truth, wherever it may be found.

D. S. JACOBUS: Dr. Keyes of the Massachusetts Institute of Technology presented a paper this fall at the *A. S. M. E.* meeting and showed that if you weigh the amount of moisture in air and then make a corresponding wet bulb measurement and use this equation, you may be in error by as much as 10 per cent. Certainly you can't expect to be closer than 2 per cent, but you may be in error by as much as 10 per cent. The perfect gas equation used in the paper may also be in error by as much as 3 per cent on the pressures considered.

W. L. FLEISHER: May I say I am glad the interest in this paper is so keen, for this reason: The Guide Publication Committee is more responsible for this paper being presented now than any other single group. We had asked Professor Goff to present a paper on theoretical thermodynamics applying to air conditioning for THE GUIDE. This paper was supposed to have been the first chapter in THE GUIDE and, when it was presented the Committee felt very definitely that it was of great importance. Since it was to supersede a chapter which had had standing in practical use for so long, the Committee didn't want to take the radical step of introducing it as the final word in the thermodynamics of air conditioning by putting it as the first chapter in THE GUIDE until it had been presented as a paper before the Society and open to discussion not only by the A. S. H. V. E. but by the physicists outside the Society. Therefore I can say as a member of the Guide Publication Committee that we welcome any criticism or any discussion of this paper, as it may well have an important bearing on future theory or future acceptance of theory by the Society.

DR. JACOBUS: I did not mean to be destructive in my criticism, but the contention of Dr. Keyes was that if it were necessary to make calculations, let us say with an accuracy of 5 per cent, certainly if you are interested in an accuracy of 1 per cent, it is essential that experimental work of a laboratory nature be made in order that an instrument might be developed which the trade could use for determining the moisture in air. This paper represents a very nice summary of the existing data. However, if we want to rely on the accuracy of our measurements, then we need another instrument or a refined wet bulb instrument, if you will put it that way, for determining the moisture in air.

J. A. GOFF: In reply to Mr. A. A. Adler's objections to the use of the term "perfect gas," I should like to point out that this term can be given precise meaning entirely consistent with the laws of thermodynamics and with experiment and is therefore perfectly legitimate. Consider the accurate experiments of Holborn and Otto (Zeit. für Physik, 23 (1924) 77, 33 (1925) 5) which showed that the isotherms of seven actual gases including hydrogen, helium, nitrogen, neon, argon, oxygen and air when plotted on the pv,p-plane and extrapolated to zero pressure give intercepts on the pv-axis in strict accord with the equation $pv = BT$. I see nothing reprehensible, therefore, in describing this limiting condition as that of a *perfect gas* with the understanding, of course, that it is a limiting condition to which all gases appear to approximate at sufficiently low pressures. Analogous reasoning is not hard to find; thus, in mechanics, we talk about a "freely falling body" knowing that in any actual case the motion is always more or less retarded by air resistance and other forces. The laws of Boyle, Gay-Lussac, Joule and Dalton must, of course, be understood to apply strictly to perfect gases only even though at the time these laws were announced they were thought to accurately characterize actual gases.

Application of the laws of thermodynamics shows that any substance obeying the simple equation $pv = BT$ must also obey the equation $h = \int c_p \cdot dT$ with c_p a function of temperature only. This conclusion is beautifully confirmed by the accurate c_p-data for water (H_2O) now available which show less and less dependence of c_p on pressure at lower and lower pressures. It is worth noting also that the limiting value of c_p at zero pressure for any temperature appears to be in strict accord with that calculated from the modern quantum theory based on the assumptions appropriate to perfect gases in which intermolecular forces of attraction are negligible.

The equation of Van der Waals must be regarded merely as an attempt to evaluate the above mentioned intermolecular forces (and the effect of finite molecular dimensions), in other words, merely as a second approximation to the laws of actual gases. The laws of thermodynamics can here again be applied to show the necessary dependence of c_p on pressure as well as temperature, and to show that this dependence on pressure must vanish at low pressures in accordance with the perfect gas laws. From a thermodynamic point of view it is important to note that actual gases which exhibit appreciable deviation from the perfect gas laws can not possibly mix in strict accordance with Dalton's Law. In other words it is thermodynamically inconsistent to use Dalton's Law in conjunction with the equation of Van der Waals. Unfortunately the extent of departure from the former in terms of the individual departures from the latter is not at present known; although it can in some cases be roughly estimated. I believe, therefore, that the use of the perfect gas laws in the present problem is justified until more accurate experimental data become available—in the interest of simplicity if for no other reason.

In reply to Dr. Jacobus, I should like to say that I am cognizant of the fact that the wet-bulb thermometer as at present constructed does not accurately indicate the moisture content of air. An important reason not generally understood is that pointed out by Arnold (Physics *4* (1933) 255, 334), namely, that the quantity of heat transferred to the wet-bulb by radiation from surroundings at dry-bulb temperature is,

even at the usual air stream velocities, 5 to 15 per cent of that transferred by convection and conduction *independently of the magnitude of the wet-bulb depression.* I have applied the radiation correction outlined by Arnold to actual tests in which a known weight of air is moved over extended surface cooled by water and have found that although the energy and weight balances computed from entering and leaving wet-bulb and dry-bulb temperatures in the conventional way (disregarding radiation) fail by large amounts; nevertheless after correction for radiation both energy and weight balances check satisfactorily. I believe it is possible to construct a wet-bulb thermometer that will have zero radiation correction and only small *calculable* corrections for partial drying of the wick, thermal capacity of the liquid, velocity of the air stream, etc. When all these corrections are either eliminated or accurately known, the readings of the instrument can be made to conform to the *thermodynamic* definition of wet-bulb temperature as given in my paper. Then, I believe, the experimental side of air conditioning will be on a much more satisfactory basis.

To Mr. W. L. Fleisher, let me reply that further study of the thermodynamics of air conditioning and classroom experience with the substance of my paper, also the valuable discussion presented above, have thoroughly convinced me of the wisdom of the Guide Publication Committee in not being too ready to accept my paper for the first chapter of THE GUIDE. I still believe, however, that the method of approach is correct and that only slight changes in presentation and addition of new material are necessary.

No. 981

THE AIR CONDITIONING SYSTEM OF THE NEW METROPOLITAN BUILDING—FIRST SUMMER'S EXPERIENCE

By W. J. McConnell, M.D.* (*NON-MEMBER*), and I. B. Kagey ** (*MEMBER*)
New York, N. Y.

THE new home office unit building of the Metropolitan Life Insurance Co. in New York City, Fig. 1, is the largest completely air conditioned office building in the world. The building is 28 stories in height, with four basements, two of which are conditioned, having a floor area of 23 acres and a capacity of 15,000,000 cu ft, the entire office sections and lunch areas being completely air conditioned.

The system was started in December, 1932 and the period of summer operation began in May and continued until late in October, 1933. It was considered undesirable to attempt to maintain any constant temperature during the summer months, in order to avoid too great a contrast between the inside and outside temperatures. The outdoor temperature on any particular day determined largely the temperature to be maintained indoors. The inside temperature approached the outside temperature as the latter approached 80 F. The maximum temperature maintained indoors reached 79 when the outdoor temperature was above 90. The average range of indoor temperature for the summer months extended between 74 and 79 F with a range of relative humidity between 43 and 53 per cent.

On entering the conditioned area during the warm weather one did not experience the sensation of chilliness, but, rather, welcomed the cool environment. One was able to work through a full-day period without the discomfort and depression formerly suffered during hot weather. It might be assumed generally that those working in a cool environment felt less fatigued at the end of the day. On leaving the building at the end of the day, the heat of the outdoor air was intense at first but the effect was not lasting.

The results of exposure of approximately 6000 employees to the conditions maintained during the summer months have been, from the viewpoint of comfort, entirely satisfactory. What effects such exposure has on the health of these employees so far have not been determined. Such health correlations

* Assistant Medical Director.
** Air Conditioning Engineer.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by Dr. W. J. McConnell.

in the past as have seemed significant are usually open to criticism for want of adequate records of conditions, or because diagnostic data were lacking to support the findings. Careful records of causes of absenteeism of all employees are made and it is hoped that an analysis of these records after a period of three or four years may assist in appraising the value of controlled air environment in relation to health.

Although dust and bacteria counts were not determined during the summer months, a dust count was made during the month of December in one of the occupied sections. The standard procedure of the United States Public Health Service, using the Greenburg-Smith impinger, was followed. The count

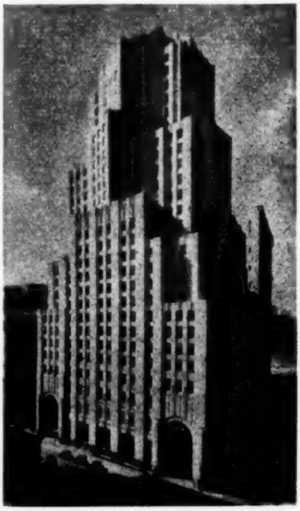


FIG. 1. AIR CONDITIONED OFFICE BUILDING OF METROPOLITAN LIFE INSURANCE CO.

amounted to 1.2 million particles per cubic foot of air in contrast to 2.9 million particles in an occupied section of an adjoining non-conditioned building. It is realized, however, that many more counts, including bacteria, should be made before any conclusions relative to the dust concentrations in the respective buildings are submitted.

While the nature of the dust encountered determines its harmfulness when inhaled or ingested, it must be remembered that dust also carries and spreads bacteria, which may be the origin of many respiratory diseases. Leonard Hill contends that the dusty, warm atmosphere of crowded rooms not only spreads infection but lessens the defense of the membrane which lines the breathing passage.

Some individuals are peculiarly sensitive to dust containing traces of foreign

material, such as pollen and molds, and suffer from hay fever, asthma, and bronchitis through breathing such dust. These individuals are greatly relieved so long as they breathe dust-free air.

That greater efficiency follows exposure of the body to environmental conditions which permit loss of excessive body heat, in contrast to a hot atmosphere, has been repeatedly demonstrated. Where the contrast is not so great, as for example, between offices not cooled during the summer months and offices maintained at a comfortable temperature, the increased efficiency of those working under comfortable conditions is difficult to measure with any degree of accuracy.

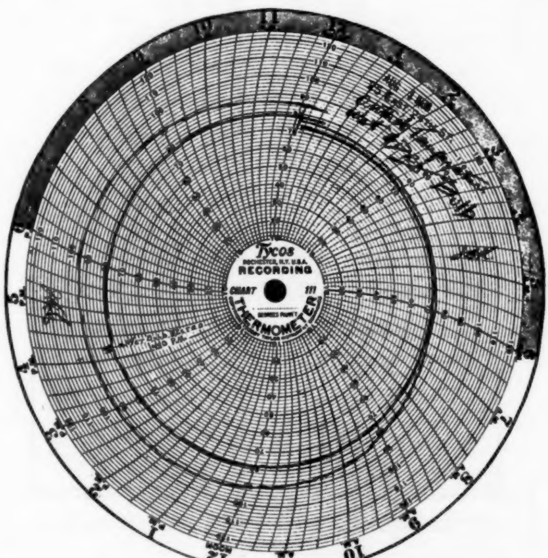


FIG. 2. TWENTY-FOUR HOUR RECORD OF WET AND DRY BULB TEMPERATURES

While the authors are convinced that efficiency is greater, no practical method of determination has been devised.

The system was operated eleven hours a day, except in extremely hot weather, when the system was started two hours instead of one hour before the office opened. Tables 1 and 2 give some of the results which were recorded during one of the hottest days in the summer of 1933. Fig. 2 gives a record of the outside wet and dry bulb temperature for a 24-hour day. Table 1 is a thermal record for the entire building between 11 a. m. and 3 p. m. Each floor that is conditioned, is divided into three exposures: north, east and south; and the average dry bulb and wet bulb temperature, with the corresponding dew point and relative humidity, is indicated for each exposure. The average of the

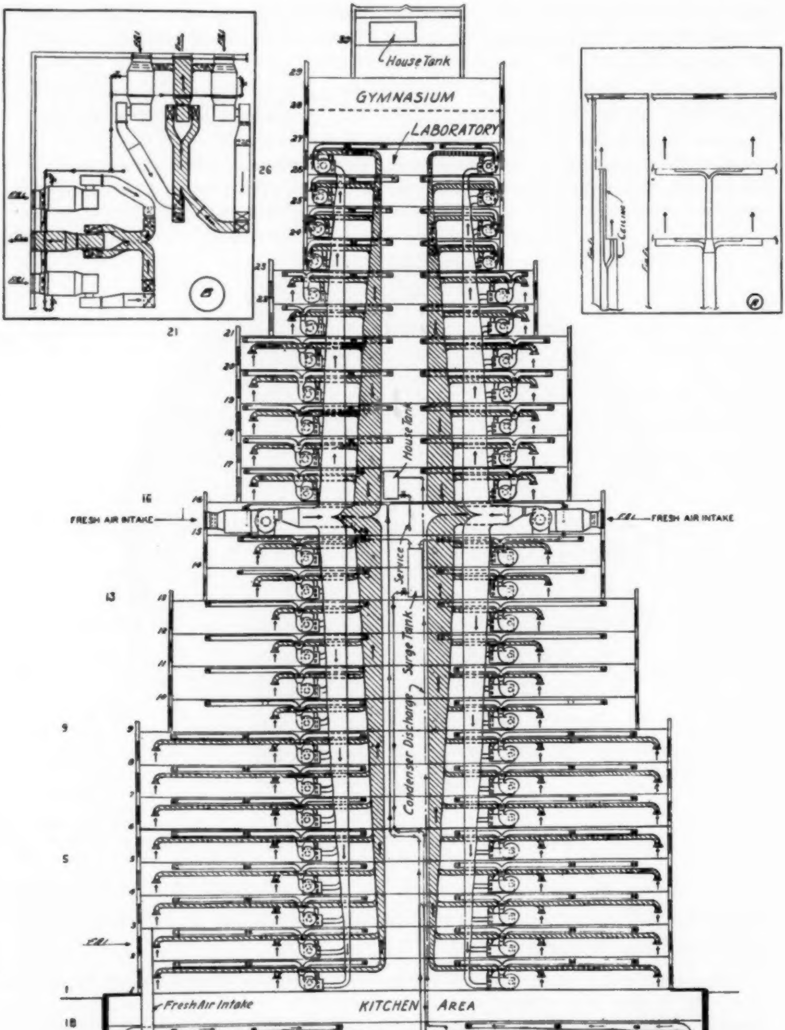


FIG. 3. DIAGRAMMATIC SKETCH OF AIR CONDITIONING PLANT

three exposures for each conditioned floor, except the 26th floor which is divided into various laboratories, is indicated under the heading average.

Table 2 is an outline of the results of a refrigeration test made at noon when the lunch rooms were occupied, thus demanding a maximum building load. On this particular day the system was started at 6 a. m., two hours before the personnel entered the building. On the day preceding, the outside temperature averaged 90 deg dry bulb and 74 deg wet bulb, and the system was shut down at 6 p. m. as usual.

Total Cooling Hours

It was found necessary to operate the refrigeration equipment for a total of 1230 hours over a period of eight months, as shown in the following tabulation:

Month	Days Refrigeration	Hours Refrigeration
April	3	31
May	16	175
June	21	226
July	20	238
August	23	273
September	20	205
October	8	62
November	3	20
Total	114	1230

If the building had operated 5½ days a week instead of 5 days it would have been necessary to operate the refrigeration for 110 additional hours or 22 half days, bringing the total up to 136 days or 1340 hours when refrigeration would be necessary.

During the 1230 hours of operation the equivalent of 29,000 tons of melting ice was produced, which would represent approximately 1,000,000 cu ft of ice.

Very definite costs as to operation, maintenance, repairs and additions are kept on the entire system. However, the data are not complete for the past year's operation due to the lack of sufficient metering equipment prior to July.

Two Main Air Conditioning Systems

The air conditioning plant is divided into two main systems, as illustrated in the diagrammatic sketch Fig. 3. One supplies the office portion of the building which consists of the first to the twenty-sixth floors inclusive, and the other supplies the lunch rooms located in the second and third subbasements. The former consists primarily of eight dehumidified air fans and four large dehumidifiers which provide air for fifty booster fans or recirculatory units. The dehumidified air fans and dehumidifiers (humidifiers in winter) are located on the fifteenth floor which is the central point of distribution (Section B-Fig. 3). The supply of outside air is received on this floor through louvers in the building's north and east walls. The outside air is mixed with a propor-

TABLE 1. METROPOLITAN LIFE INSURANCE COMPANY — FLOOR TEMPERATURE AND HUMIDITY — AUGUST 1, 1933

Air Conditioned Space	North				East				South				Average			
	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%
1st Floor........	76.2	63.3	55	49	76	63	55	48	80	65.2	56.5	45	77.4	63.8	55.5	47.3
1st Mez..........	77	63.5	55	47	77	63.5	55	47	76.5	63.5	55.2	48	76.8	63.5	55	47.3
2nd Floor........	77	63.5	55	47	78.5	64.5	56	46	77.5	65	57	50	77.7	64.3	56	47
2nd Mez..........	76.5	63.5	55.2	48	77.8	64	55.6	46	78.5	64.8	54.8	44	77.6	64.1	55.2	46
3rd Floor........	77.5	64.3	56.5	48	76.8	63.5	55	48	76	63.2	55.2	49	76.8	63.7	55.6	48.3
3rd Mez..........	76.5	63.5	55.2	48	77.3	65	57	50	76.8	63.8	55.5	48	76.8	64.1	55.9	48.7
4th Floor........	77.5	63.5	55	45	79	65	56.5	46	76.2	62.8	54.5	46	77.6	63.8	55.3	45.7
4th Mez..........	76.5	63.5	55.2	48	77.8	64	55.6	46	77	63.5	55	47	77.1	64.5	55.3	47.7
5th Floor........	78	64	55.5	46	79	65	56.5	46	78.8	64.2	55	44	78.6	64.4	55.9	45.3
6th Floor........	79	65	56.5	46	78.5	64.5	55.3	44	76.8	63.2	54.8	47	78.1	64.2	55.6	45.7
7th Floor........	79	65	56.5	46	78.2	63.8	55	44	76	61.5	52.5	46	77.4	63.4	54.9	45.3
8th Floor........	79.5	65.5	57.2	47	77.5	63.4	54.6	45	77.2	63.5	55.2	47	78.1	64.1	55.7	46.3
9th Floor........					78.5	64.5	56	46	78.5	64.5	56	46	78.5	64.5	56	46
10th Floor.......	78.6	65.3	57.5	49	77.5	64	55.2	46	77.5	63.8	55.2	46	77.9	64.4	55.4	47
11th Floor.......	78.5	65	57	48	78.5	64.5	56	46	78.5	64.5	56	46	78.5	64.7	56.3	46.9
12th Floor.......	77	64	55.5	44	77.5	63.7	55.2	46	78.5	64	55.5	44	78.2	63.9	55.4	44.7
13th Floor.......	77	63.5	55	47	76.5	63.5	55.2	48	78.5	64.5	56	46	77.3	63.8	55.4	47

Readings by Operating Staff, Metropolitan Life Ins. Co., 11 a. m. to 3 p. m.

tionate amount of air returned from the various floors and in turn passes through the dehumidifiers where it is properly conditioned. After leaving the dehumidifiers in an almost saturated state, the air is forced to the upper and lower floors through galvanized iron shafts which are cork insulated. There are four main supply shafts, two of which supply the northerly and two the southerly halves of the office areas of the building with conditioned air. There are two booster or recirculatory fan units located in each of the conditioned floors which derive their supply of conditioned air from the north and south main shafts, respectively. Each unit draws a proportionate amount of air from the office area it supplies and this air, in turn, after mixing with the conditioned air supplied to the unit, is delivered to the office area. During the winter the humidified air is supplied from the shaft and after mixing with the air drawn from the office area is passed through small booster heating coils which temper the air to 70 F before it is discharged into the office area. The heat losses in the office areas are offset by means of radiators located under each window.

TABLE 1. (*Continued*)

AIR CONDITIONED SPACE	North				East				South				Average			
	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%
14th Floor	78	63.5	55	47	77.5	63.8	55.2	46	80.2	65.2	66.3	44	78.2	64.2	55.5	45.7
16th Floor	79	62.5	53	42	78.5	63.5	54	43	78	63	53.5	43	78.2	63	53.3	42.7
17th Floor	78	64	55	43	79	64	55	43	79	64	55	43	79	64	55	43
18th Floor	76	63	63.5	43	78	63	53.5	43	77.5	63	54	44	77.8	63	53.5	43.3
19th Floor	78	62.5	54	46	77	63	54	45	77.5	63.5	55	45	76.8	63	54.3	45.3
20th Floor	76	63.5	55	45	78	64.5	56.5	48	79	65	56.5	46	78.3	64.3	55.7	46.3
21st Floor	77	62	53	45	77	63	54	45	76.5	63	55	47	76.5	62.7	54	45.7
22nd Floor	79	63	54	45	79	63.5	54.5	42	77.5	63	54	44	77.8	63.2	54.2	43.7
23rd Floor	79	64	55	43	78	63	53.5	43	77	63	54	45	78	63.3	54.2	43.7
24th Floor	78	64	55	43	78.3	63.3	54	43	78	63.5	54	44	78.4	63.6	54.5	43.3
25th Floor		63.5	54	44	78	64	53.5	46	78.5	63.5	54	43	78.2	63.7	53.8	44.3
26th Floor 1									79	62.5	51.5	39				
26th Floor 2									76.5	62	52.5	43				
26th Floor 3					74.5	63	56	52								
26th Floor 4					77	63.5	55.5	47								
26th Floor 5	77	63	53.5	44												
26th Floor 6	78.5	66	59	52												
2nd Basement	77	66	59.5	56	77.5	66	59	54	77	66	56.5	56	77.2	66	59.3	55.3
3rd Basement					78	65	57.5	50	80	67	59.5	50	79	66	58	50

Combination return and exhaust fan (Fig. 3, sketch B) are located on the fifteenth floor and are connected at this level with four return shafts installed alongside the four main supply shafts. These fans draw from the shafts air exhausted from each floor and either deliver it to the dehumidifier or exhaust it to the outside.

METHOD OF INTRODUCING AIR

The method of introducing the conditioned air into the office areas is of interest because the method adapts itself both to architectural appearance and to uniform distribution. The office areas generally are large open spaces having very few partitions as indicated in Fig. 5. The large surface of ceiling is divided into successive steps which radiate upward and outward from the service area of the floors to the windows on three sides of the building. The average ceiling has four such steps each about 7 in. above the preceding step. The openings from the supply outlets are located on the face of these

steps and concealed by specially designed strip metal grilles 4 ft in length and 4½ in. in height. (Fig. 3 sketch A.)

Careful study was given to the design of the ducts concealed in the ceiling and leading from the two respective fan rooms to the supply grilles in order to maintain maximum ceiling height. The duct system consists of numerous branches having an average depth of 7 in. and in some instances these branch ducts pass through openings cut out in the steel girders to permit the ducts to

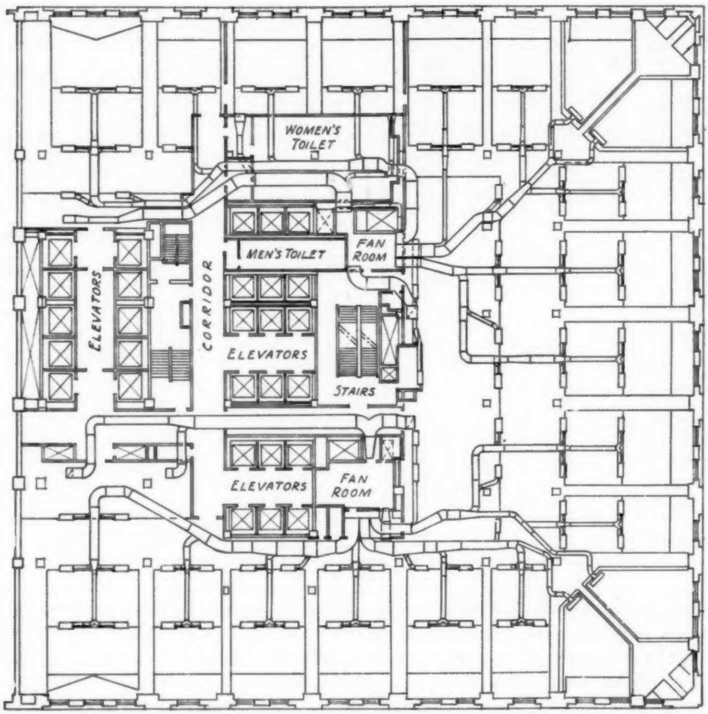


FIG. 4. FLOOR PLAN SHOWING TYPICAL DUCT LAYOUT

reach extreme corners of the office areas. A typical floor plan showing the duct layout is shown in Fig. 4.

The air is exhausted from the office areas through large registers located in the ceiling soffit. There are on an average ten return registers per floor and as there are few partitions, these registers are situated adjacent to the inside service walls, thus eliminating the necessity of an extensive return duct system which would occupy valuable head room.

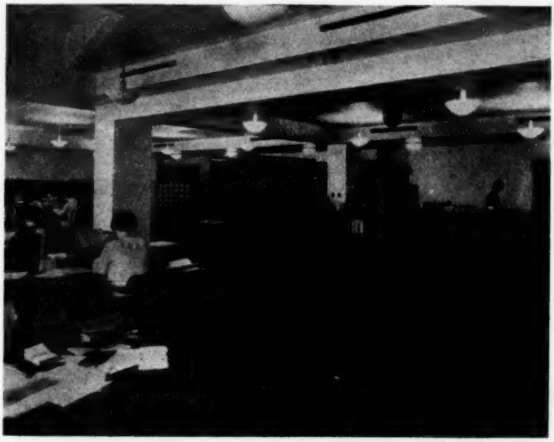


FIG. 5. AN OFFICE AREA SHOWING METHOD OF AIR DISTRIBUTION

The dehumidifying equipment for the lunch rooms is located in the fourth subbasement and draws its supply of outdoor air through a shaft leading to stone grilles in the north wall at the second floor level.

This system, which is the second of two main systems, consists of three dehumidifiers, two of which supply the lunch room in the second subbasement and the third supplies the lunch room in the third subbasement. Unlike the first main system previously described, this system recirculates and returns the air at each dehumidifier thus requiring only three fans.

TABLE 2. METROPOLITAN LIFE INSURANCE COMPANY — REFRIGERATION CAPACITY TEST

August 1, 1933
Outside temp. 91 deg. D.B. — 78.5 deg. W.B. Four machines full speed

Water entering condensers	66 F
Water leaving condensers	87 F
Water entering coolers	49.5 F
Water leaving coolers	40.0 F
Temp. drop in coolers	9.5 F
Gpm through coolers	3420
Heat absorbed per hr	16245000 Btu
Tons of refrigeration	1353.7
Tons of refrigeration to office space	1125.7
Tons of refrigeration to restaurants	228.0
Kw in-put	1222
Hp in-put	1635
Hp in-put per ton	1.21
Gpm condenser water	1880
Gpm condenser water per ton	1.39

FIG. 6. LUNCH ROOM AREA SHOWING AIR DISTRIBUTION

The distribution of air to these areas is effected in a different way from the method used in the office areas on account of the different type of ceiling. In these areas the ceilings are flat and the air is introduced into the rooms through one foot square openings in the ceiling. Suspended directly under these openings and four inches below the ceiling are solid plates three feet square which serve as baffles for the distribution of air as shown by Fig. 6.

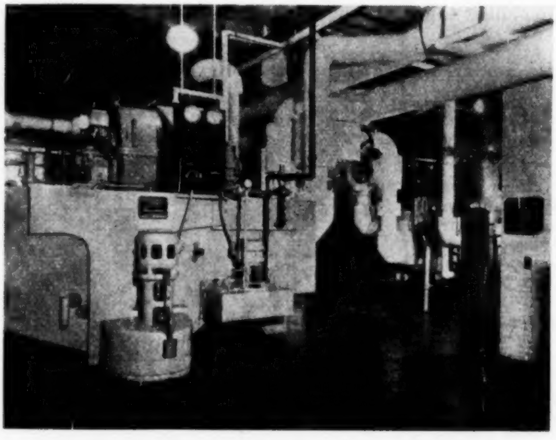


FIG. 7. VIEW OF REFRIGERATION PLANT

The air is returned through registers located around the walls and columns. The duct work in these areas conforms with standard practice.

REFRIGERATION EQUIPMENT

The refrigeration equipment (Fig. 7) for the entire air conditioning system consists of four centrifugal machines each driven by a 350 hp variable speed motor. These machines, each weighing 30 tons, are located in the fourth subbasement. Three centrifugal pumps (1800 gpm) are located adjacent to the refrigeration machines. Two of these pumps carry the entire load, the third is an auxiliary. The pumps and intercoolers are shown in Fig. 8. The water to be cooled is forced by these pumps through the coolers of the refrigeration machines into a 12-in. all welded pipe system which carries the water to the fifteenth floor where the piping system branches into four 8-in. lines, each of

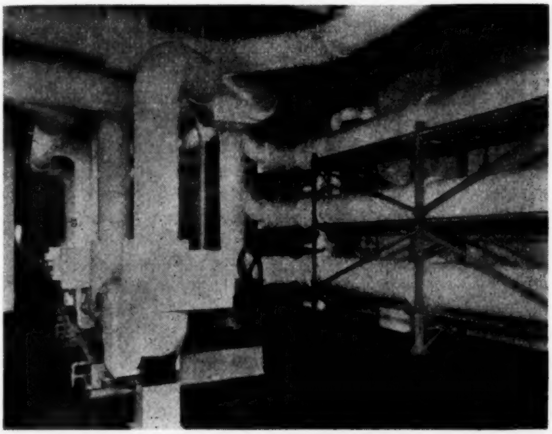


FIG. 8. PUMPS AND INTERCOOLERS

which, in turn, supplies a booster pump. These booster pumps are located at each of the four dehumidifiers. The water is forced through the sprays of the dehumidifiers by the booster pumps and is carried by gravity through individual lines leading from the collecting pans of the dehumidifiers to an 11,000 gal surge tank located on the thirteenth floor. The water returns from the surge tank to the suction side of the pumps located in the fourth subbasement through a 12-in. pipe line to complete the refrigerated water circulating circuit.

The water for the system supplying the lunch rooms is chilled by means of three intercoolers located in the refrigeration room and inserted into the cold water supply line leading from the refrigeration machines to the fifteenth floor. Each subbasement dehumidifier is connected with an independent inter-

cooler. The water to be chilled is passed through the intercooler prior to forcing it through the sprays.

The following is a summary of the magnitude of the system:

Air supplied	540,000 cfm
Square feet floor area conditioned	650,000
Refrigeration	1,350 tons
Cold water circulated	3,400 gpm
People	6,500
Seating capacity restaurants	2,500
Lunches served per day	7,800
Electrical power connected	2,170 hp
Fans	71
Dehumidifiers	7
Pumps	10

No. 982

OBSERVATIONS OF HAY-FEVER SUFFERERS IN AIR-CONDITIONED ROOM AND THE RELATIONSHIP BETWEEN THE POLLEN CONTENT OF OUTDOOR AIR AND WEATHER CONDITIONS

By T. A. Kendall * and Garland Weidner,† M.D., C.P.H. (*NON-MEMBER*)
Lexington, Ky.

IN the spring of 1932 a room known as Dicker Hall, located in the mechanical engineering building on the University of Kentucky campus, was equipped with a unit air-conditioner. Dicker Hall is the social club room for the students of the College of Engineering and is normally used for that purpose only, but at the beginning of the hay-fever season of 1932, through announcements in local newspapers, the room was made available to the public as offering a possible haven for hay-fever sufferers. The invitation met with a hearty response. No records of the visitors were obtained during this season but many visitors reported informally the relief of some of their symptoms.

At about the same time, due to interest in air filters for filtering pollinated air, a study of the pollen content of outdoor air was begun. Slides coated with corn oil were exposed daily outdoors in front of the engineering building (Station A). Metal slide shelters were used similar to those described by Gay.[1] The slides were exposed each morning between 8 and 9 o'clock and were allowed to remain for a period of 24 hours.

In making the counts, a Bausch and Lomb microscope, having a standard 16 millimeter objective and a 10X ocular, was used. An area of 113 square millimeters was observed by three trips across the slide. These counts were multiplied by a factor (0.885) giving the number of pollen per square centimeter. A plot of the daily counts for the 1932 season is shown in Fig. 1.

So far as making any practical use of the 1932 counts, the principal interest settled around the relationship between pollen counts and weather conditions. Meteorological data were obtained from the local weather bureau and were

The research work reported on in this paper was conducted in the Johnston Solar Laboratory of the College of Engineering, University of Kentucky, under the general direction of Dean F. Paul Anderson and in cooperation with the University of Kentucky Department of Hygiene.

* Research Assistant, College of Engineering, University of Kentucky.
† Department of Hygiene, University of Kentucky.

[1] A Survey of the Pollen Flora in Baltimore During 1929, by Paul Acquarone and Leslie N. Gay, *Journal of Allergy:* Vol. II, 1931, p. 336.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by W. H. Driscoll.

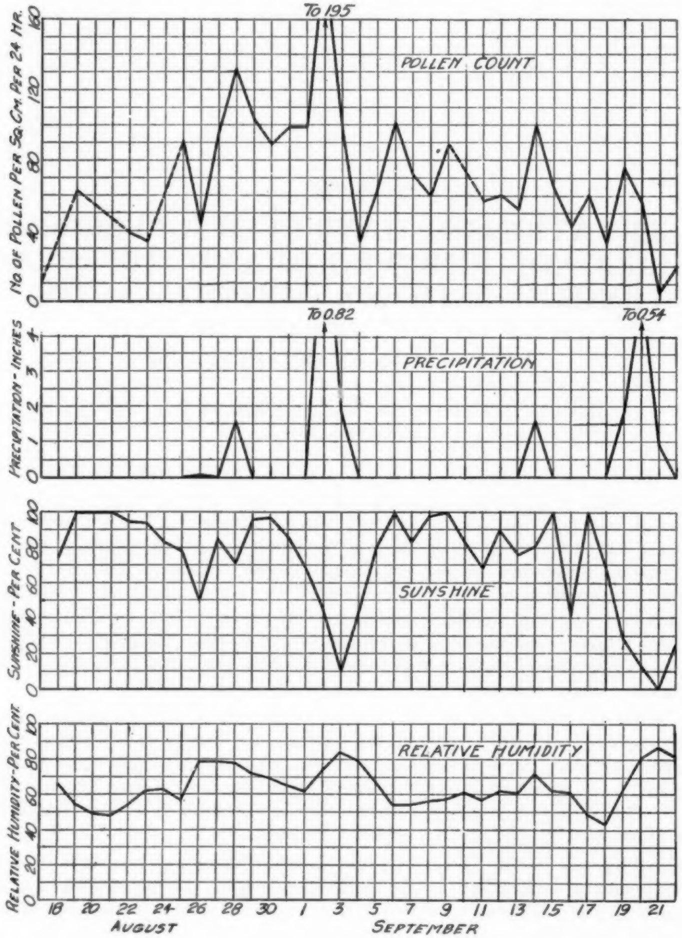


FIG. 1. DAILY POLLEN COUNT AND WEATHER CONDITIONS DURING THE RAGWEED SEASON, LEXINGTON, KY., 1932

tabulated for each day as shown in Table 1. Plots of precipitation in inches, percentage of sunshine and relative humidity are shown in Fig. 1.

In 1926, Ray M. Balyeat, M.D., made a study of the pollen content of the air and weather conditions in Oklahoma City.[2] The relationship between precipitation and pollen count is stated by Dr. Balyeat as follows:

[2] Allergic Diseases, Their Diagnosis and Treatment, by Ray M. Balyeat, p. 99, F. A. Davis and Co., 1930.

The amount of precipitation plays a very definite part in several ways in determining the amount of pollen that actually remains in the air. In the first place, the amount and distribution of rainfall during the growth of plants determine largely the number of plants and the extent of their growth. They also determine the amount of pollen that will be produced by the plant life, inasmuch as large stalks of Bermuda or ragweed will produce more pollen than small ones.

After the period of anthesis arrives, Dr. Balyeat apparently considers the washing of pollen from the air as the only effect of precipitation upon the pollen content of the air. On the day following a heavy rain in Oklahoma City, the pollen count reached one of the highest peaks of the season. Dr. Balyeat says:

On the day following the heavy precipitation first mentioned, the wind began to blow about 3 o'clock in the morning and continued throughout the day. The percentage of sunshine on the following day was 100. Under such conditions the precipitation had practically nothing to do with regulating the amount of pollen in the air.

It is the opinion of the writers however, that this high count was due to the effect of precipitation as well as sunshine upon plant life, for the reason that a rain causes the plants to grow more rapidly and produce an increasing amount of pollen to be scattered during the dry period following the rain. For example, the high counts obtained on August 28 and September 2, 14 and 19, as shown in Fig. 1, occurred on days preceded by rainfall. This was true for September 2, in spite of the fact that the percentage of total possible sunshine was very low on this day. The total precipitation shown in Fig. 1 was for the same period covered by the pollen count but the precipitation curve does not show the part of the 24-hour period during which the precipitation occurred; the high precipitation and high pollen count for some days appear to occur simultaneously, and hence the foregoing statement does not seem to be substantiated by the data as plotted. This can be explained by reference, for instance, to the complete data for September 2. The precipitation on this day fell between 11:00 a. m. and 2:00 p. m. and was followed by bright sunshine; while the pollen count obtained for the same date was for the 24-hour period ending about 9 o'clock the following morning.

Dr. Balyeat states that a high percentage of sunshine causes a high count. The pollen counts of August 29, 30, and 31, and on September 6 were probably above normal due to the high percentage of sunshine on those days. For a day or so following a peak in the pollen count, the count is usually low, regardless of weather conditions. This is due probably to a shortage of matured anthers caused by the previous heavy yield. For example, the low count on September 4 may have been caused by the heavy yield which occurred on September 2.

Relative to wind movement, Dr. Balyeat says:

The velocity of wind, to a considerable extent, determines the amount of pollen that gets into the air, how far it will be carried, and the amount that comes in contact with the nasal mucous membrane of the hay fever sufferer. The time of day in which the wind blows is important. For example, if there is a stiff gale throughout the early morning hours, during which time the pollination is the heaviest, it will carry more pollen into the air than a similar gale during the evening.[3]

The average velocity in Lexington during the 1932 season was 9.89 mph. There were five days with an average velocity above 12 mph and three days

[3] Allergic Diseases, Their Diagnosis and Treatment, by Ray M. Balyeat, p. 102, F. A. Davis and Co., 1930.

TABLE 1. TABULATION OF DATA — 1932

Date	No. of Pollen Per Sq Cm Per 24 Hours	Dry Bulb Temp. F	Relative Humidity Per Cent	Sunshine Per Cent	Precipitation Inches	Average Wind Velocity Miles per Hour							
						North	North-East	East	South-East	South	South-West	West	North-West
Aug.													
17	11	...	...	...	...	...	...	...	...	...	...	...	...
18	...	70.1	66	75	T	6.16	.83	...	...	...	...	.50	3.91
19	63	66.5	54	100	...	3.75	4.25	4.04	...	...	...	...	...
20	...	67.9	49	100	...	...	5.04	8.66	...	...	...	...	...
21	...	71.3	48	100	...	...	1.75	10.25	...	...	...	...	...
22	39	72.9	54	95	...	1.25	.46	8.25	...	...	...	...	...
23	34	74.6	62	94	...	...	.58	2.26	2.83	2.21	...	...	...
24	...	75.4	63	83	...	...	...	...	...	5.25	3.92	.67	...
25	91	80.6	57	78	...	...	...	...	...	5.79	5.66	.96	...
26	43	73.0	79	50	.01	...	...	...	...	...	11.09	2.00	...
27	95	76.5	79	85	...	...	.13	.25	.25	...	.83	1.92	3.92
28	132	76.6	78	71	.16	...	...	...	.38	5.75	1.75	1.17	.46
29	103	80.0	72	96	T	...	...	...	...	.25	3.96	5.00	...
30	89	80.2	69	97	T	.38	...	...	...	...	1.88	2.08	.83
31	99	82.8	65	86	...	...	5.41	.79	1.79	.96	...	...	...
Sept.													
1	99	83.5	62	69	T	...	1.17	3.46	3.62	2.00	...	...	...
2	195	79.4	74	45	.82	2.33	.25	4.66	.92	...	.92	...	...
3	96	74.7	84	10	.18	...	...	...	1.46	4.42	5.87	3.92	...
4	34	75.5	79	43	...	1.50	.25	.21	...	...	4.67	1.21	.25
5	63	72.9	67	81	...	4.42	...	.13	...	...	...	...	4.75
6	102	64.1	54	100	...	9.34	2.79	...	...	...	...	...	...
7	72	61.4	54	83	...	5.38	6.70	.38	...	...	...	...	...
8	60	63.9	56	98	...	4.16	4.54	4.34	...	...	...	...	...
9	89	70.6	57	100	...	...	...	10.91	...	.88	...	...	...
10	...	72.8	61	83	T	...	1.08	4.17	2.38	1.75	...	...	...
11	57	73.6	57	68	...	.25	1.96	3.21	2.34	.29	...	.50	.79
12	60	73.0	62	90	...	...	...	1.08	1.50	2.54	4.96	...	...
13	52	73.9	61	76	...	.29	1.00	.54	.88	1.46	2.21	2.04	...
14	100	71.5	72	81	.16	1.58	1.67	2.75	.25	.21	.25	.33	.21
15	64	72.6	62	100	...	6.21	.88	...	...	...	...	...	1.38
16	42	66.0	61	41	...	1.54	...	...	...	...	...	.54	6.66
17	60	64.6	48	100	...	...	...	...	2.88	.46	.21	...	4.84
18	33	73.8	43	69	...	...	...	...	6.54	7.87	...	...	...
19	76	73.5	63	28	.18	...	...	2.29	.46	6.25	3.00	...	...
20	56	67.1	81	13	.54	...	...	2.62	4.59	3.71	4.29	.38	...
21	5	66.2	87	0	.09	...	...	.63	1.92	1.25	.25	2.21	1.71
22	19	69.3	81	25	T	6.75	.50	.92	...	.50	...	...	...
23	1	61.8	69	29	...	3.12	10.21	...	...	...	...	...	...
24	24	59.4	57	100	...	.54	10.18	3.12	...	...	...	...	...
25	19	65.2	55	62	.09	...	...	9.00	4.00	1.00	...	...	...
26	4	63.9	87	0	.60	...	...	...	...	12.39	.50	...	...
27	4	64.7	86	6	.10	.88	...	...	...	1.75	7.38	...	3.21
28	2	56.9	65	98	...	2.00	...	...	...	...	...	1.13	6.30
29	3	58.6	56	55	...	2.46	3.71	3.12	.79	...	...	.58	1.25
30	0	59.0	60	97	...	...	...	.87	.37	.46	.25	.25	3.20

with a velocity below 8 mph. A study of the data has failed to reveal any correlation between wind velocity, direction of wind and pollen count. The effect of wind was probably out-weighed by other weather influences. Also it is probable that no correlation between these factors could have been obtained

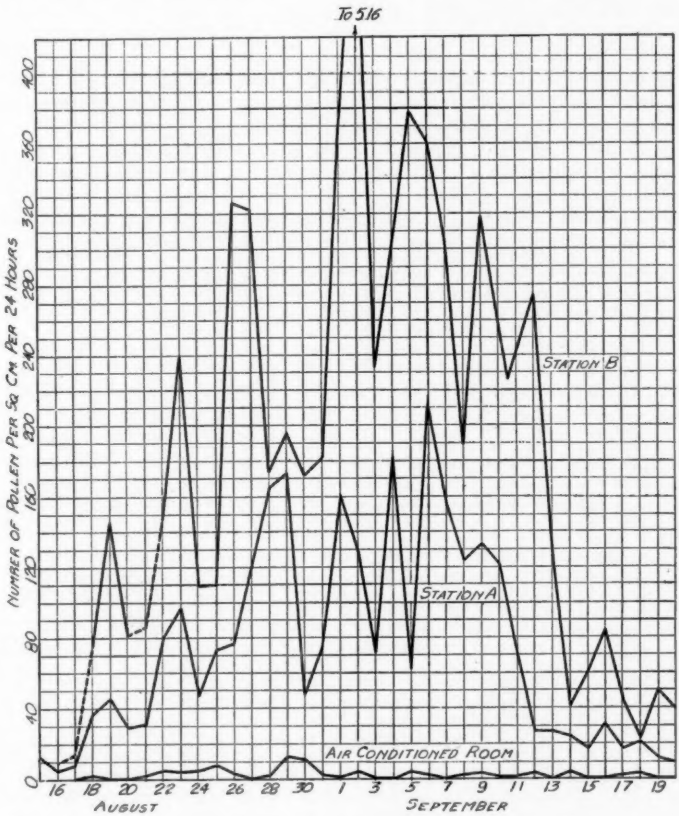


FIG. 2. DAILY POLLEN COUNT AND WEATHER CONDITIONS DURING THE RAGWEED SEASON, LEXINGTON, KY., 1933

even though other weather conditions had remained constant, for the reason that the sources of the pollen were fairly well distributed around the station.

The relationship between the pollen content of the air and weather conditions may be summarized by stating the following general conclusions which appear to be substantiated by the data:

(1) The effect of rain is twofold: (a) rain washes the air practically free of pollen; and (b) rain accelerates the growth of the plant and causes a high yield of pollen

TABLE 2. TABULATION OF DATA — 1933

Date	Number of Pollen Per Sq Cm Per 24 Hours			Temperature and Humidity: Outside 24 Hours		Temperature and Humidity: Outside 9 A.M. to 9 P.M.			Temperature and Humidity: Air Conditioned Room 9 A.M. to 9 P.M.			Precipitation	
	Station A	Station B	Air Cond. Room	D. B. T. F	Per Cent Rel. Hum.	D. B. T. F	Eff. T. F	Per Cent Rel. Hum.	D. B. T. F	Eff. T. F	Per Cent Rel. Hum.	Inches	Time of Day
Aug.													
15	13	..	..	71.5	62	75.0	68.6	53	69.8	66.0	66	..	....
16	5	9	..	74.0	67	78.9	72.6	58	72.5	69.0	72	..	....
17	8	..	0	71.0	83	74.0	71.0	79	74.2	70.5	73	.10	Noon
18	37	77	2	74.6	71	80.0	75.2	67	74.4	70.4	70	T	....
19	45	146	0	73.1	67	77.7	71.7	59	74.9	70.9	70	...	....
20	29	81	0	75.0	67	79.2	72.8	58	74.8	70.9	70	...	....
21	31	...	2	76.0	73	81.0	75.6	67	75.6	71.6	71	...	....
22	80	151	5	74.4	71	78.0	72.7	64	74.8	70.7	70	...	....
23	97	240	4	75.6	68	81.0	74.1	58	75.0	71.0	71	...	....
24	47	109	5	73.8	74	77.7	72.6	67	74.5	70.9	74	...	....
25	73	110	8	76.6	68	81.6	74.8	59	76.6	72.8	72	...	....
26	77	326	3	79.7	61	84.0	75.8	52	75.9	71.5	69	...	....
27	119	322	0	79.0	65	85.0	76.1	49	76.6	72.5	71	.01	Last hour
28	165	174	2	67.4	84	67.7	66.5	90	72.1	69.4	76	.35	First morning
29	173	196	13	70.8	65	74.1	68.3	57	72.0	68.6	74	.02	Last morning
30	47	172	11	69.8	84	69.8	67.5	80	71.0	68.5	78	.57	Both mornings
31	75	182	2	71.7	84	74.6	72.0	81	75.5	72.5	78	.05	First morning
Sept.													
1	161	368	1	74.5	78	75.2	71.6	75	76.1	72.8	76	...	....
2	129	516	4	75.1	86	76.5	74.0	83	75.8	72.9	79	.12	Afternoon
3	71	233	0	74.1	88	77.3	74.8	83	74.6	72.0	81	1.74	Evening
4	182	304	0	74.5	90	75.9	74.5	88	76.5	74.0	82	...	....
5	62	378	4	75.8	76	81.2	76.1	69	77.0	73.9	77	...	....
6	212	361	2	75.9	71	80.5	74.5	64	77.4	73.5	73	...	....
7	156	304	0	78.0	72	82.7	76.3	63	76.3	72.4	72	...	....
8	124	189	2	79.7	76	84.1	78.3	67	75.5	71.9	73	...	....
9	133	318	3	81.2	72	85.0	78.9	67	76.6	72.7	73	...	....
10	121	...	1	79.4	71	82.7	76.8	64	76.4	72.2	70	T	....
11	71	[a] 22	61	79.4	74	85.0	78.3	62	76.9	72.9	72	.58	Last morning
12	27	274	3	75.0	85	78.3	75.2	80	74.6	71.4	76	.56	During day
13	27	126	0	75.0	86	76.7	74.1	83	74.9	71.5	75	.65	During night
14	24	41	4	76.4	85	80.5	76.6	77	75.1	71.6	75	.29	Afternoon
15	17	60	0	75.0	86	78.1	75.1	80	76.5	73.1	76	...	....
16	31	84	0	78.2	80	80.4	76.3	75	76.6	73.2	76	...	....
17	17	44	2	75.3	60	80.9	73.6	54	77.4	72.9	70	...	....
18	21	23	3	70.6	61	75.3	68.6	53	77.3	72.0	64	...	....
19	12	50	0	77.8	63	82.2	76.0	61	76.5	72.1	69	...	....
20	9	39	0	64.5	48	70.1	63.6	42	73.7	68.7	61	...	....

[a] One half of count for 48 hours.

following the rain. (2) A high percentage of sunshine will increase the pollen yield due to the ripening and opening of matured anthers. (3) Following a peak in the pollen count the count is usually low for a day or so, regardless of weather conditions, because of a shortage of matured anthers due to the previous heavy yield. The mag-

nitude of the count will, of course, vary with the seasonal trend, being relatively low at the beginning and end of the season and relatively high during the middle of the season.

When the 1933 hay fever season approached it was decided to make Dicker Hall available again for hay fever sufferers; and this time, to observe the visitors and make a record of the relief they obtained. Also, the pollen count of the outdoor air was continued, and in addition a record of the pollen content of the air in Dicker Hall was made, supplemented by a record of the room temperature and relative humidity. Outside weather conditions were observed also, but due to a curtailment of the activities of the weather bureau complete meteorological data were not obtained. Another outdoor station (Station B), located at the residence of one of the writers, about a mile and one-half from the university campus, was established to obtain a check count of the pollen content of the air and to observe the difference due to location. Table 2 shows the indoor and outdoor pollen counts, temperatures and humidities, and precipitation for each day of the 1933 season. A plot of the pollen counts for Stations A and B, and for the air conditioned room is shown in Fig. 2.

Comparing the outdoor counts for the 1932 and 1933 seasons, it will be noticed that the distribution for the 1932 season was fairly uniform from August 19 to September 20, while the count for 1933 shows a decided upward trend until the peak was reached near the middle of the season and then a more precipitous but yet gradual and fairly even decline to the minimum at the end of the season. This difference in the distribution of pollen for the two seasons may be accounted for by the fact that during the 1933 season practically all of the precipitation came within one period of seven days, from August 28 to September 3, while in 1932 there were four separate rains occurring at intervals during the season with nearly clear days between. It will be noticed that the count at Station B was much higher than at Station A during the 1933 season. This was probably caused by the source of the pollen being nearer Station B than Station A, and perhaps also to some relative difference in the degree of exposure of the slides.

The unit air-conditioner installed in Dicker Hall was of conventional design as shown in Fig. 3. It consisted of a three wheel fan with separate outlets, a heater, and two banks of sprays. The return air entered near the top of the unit at the rear, traveled downward parallel to one spray, then upward on the front side counter to the other spray, through the heater, fan and discharge outlets. The water for the sprays was cooled by means of refrigeration equipment already installed in connection with the Johnston Solar Laboratory which is adjacent to Dicker Hall.

The control of the unit was entirely by hand, through the regulation of the quantity of spray water used, the manipulation of a mixing valve which regulated the relative amount of refrigerated water and recirculated water, and through the adjustment of the fan speed by means of a four-step electric controller. The heater was not used at any time.

There were a few days when the outside temperature was relatively low, and it was necessary to run with a relatively high spray water temperature to avoid cooling the room below a comfortable temperature, and sometimes to cut off the spray water entirely to avoid increasing the relative humidity. To

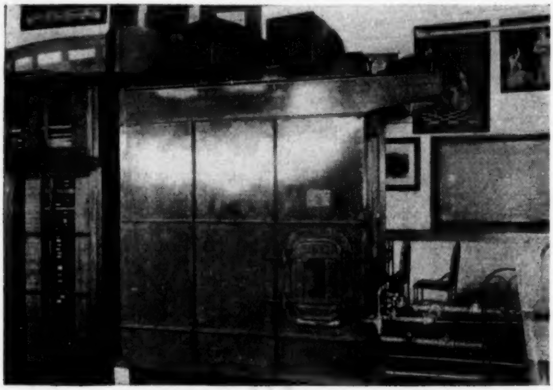


FIG. 3. FRONT VIEW OF AIR CONDITIONING UNIT

provide for cleaning the air on days when the spray water was cut off, a bank of filters was placed before the inlet to the air-conditioner as shown in Fig. 4, and the fan was operated to circulate the air through the filters, and to produce a circulation of the air within the room itself. Plots of effective temperature and relative humidity of the indoor and outdoor air are shown in Fig. 5. The values shown for each day are the average values for only the part of the day that the room was occupied, that is, for the period from 9:00 a. m. to 9:00 p. m.

The unit air-conditioner handled approximately 2000 cu ft of air per minute (at one of the lower fan speeds at which the unit was usually operated), creating about two and one-half air changes per hour by recirculation only. The air motion at the breathing line in Dicker Hall as determined by a Kata

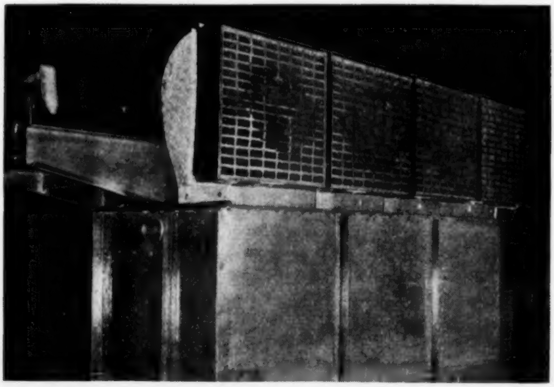


FIG. 4. REAR VIEW OF AIR CONDITIONING UNIT

thermometer ran from 12 fpm to 41 fpm, averaging about 25 fpm. The effective temperatures shown in the table have been determined from the wet and dry bulb temperatures on the assumption of still air conditions only, the influence of air movement having been disregarded.

Before describing the result of observing the relief of hay fever symptoms in the air-conditioned room, some consideration should be given to the peculiarities of the disease called hay fever and to the experience which the medical profession has had with this disease.

As early as 1819 Bostock believed that hay fever was caused by "emanations from hay." As long ago as the middle of the 19th century Blackley noted that there were marked variations in the pollen content of the air during the

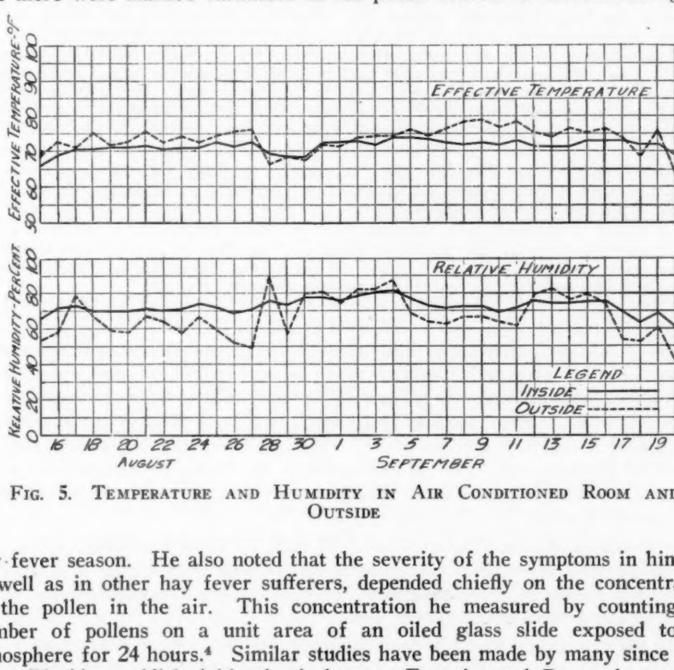


FIG. 5. TEMPERATURE AND HUMIDITY IN AIR CONDITIONED ROOM AND OUTSIDE

hay fever season. He also noted that the severity of the symptoms in himself, as well as in other hay fever sufferers, depended chiefly on the concentration of the pollen in the air. This concentration he measured by counting the number of pollens on a unit area of an oiled glass slide exposed to the atmosphere for 24 hours.[4] Similar studies have been made by many since 1873 when Blackley published his classical paper Experimental Researches on the Causes and Nature of Catarrhus Aestivus.

Duke and Durham in 1928 wrote that, "The marked irregularity of the charts (pollen) often accounts for variation in the symptoms displayed by patients from day to day during a given season."[5] Rackemann and Smith of Boston[6] and Acquarone and Gay of Johns Hopkins have shown graphically and con-

[4] Experimental Researches on the Causes and Nature of Catarrhus Aestivus, by Chas H. Blackley, London, 1873.

[5] Pollen Content of Air; Relationship to Symptoms and Treatment of Hay Fever, by W. W. Duke and O. C. Durham, *Journal of American Medical Assn.* 90:1529, 1928.

[6] Studies in Asthma XII, The Plant Pollens of New England, by Francis M. Rackemann and Lyman B. Smith, *Journal of Allergy*, Vol. II, 1931, p. 364.

vincingly this correlation between severity of symptoms and the number of grains of pollen in the atmosphere to which the patient was exposed.[7]

An illuminating historical sketch is given by Thommer as follows:

When in 1819 Bostock published his now famous paper, Case of Periodical Affection of the Eye and Chest, he had already been a sufferer from the malady, later known as hay fever, for 38 years. Concerning his own treatment he tells us: "Topical bleeding, purging blisters, spare diet, bark and various other tonics, steel, opium, alterative courses of mercury, cold bathing, digitalis, and a number of topical applications to the eyes, have been fully tried." Though these measures were "persevered in with an unusual degree of steadiness" he informs us that "it is doubtful whether any distinct or permanent benefit has been derived from any of it ever."

It is apparent, therefore, that from the very outset, the disease must have been recognized as being inherently intractable. Bostock's estimate of the efficacy of the various therapeutic endeavors in his own case, is abundantly substantiated by the subsequent history of the treatment of the malady, for we find that during the next hundred years the manifold measures and multitudinous medicaments advocated as cures and as means of obtaining relief bear testimony by their very number to their general inefficiency and unsatisfactory character. The lack of adequate progress, notwithstanding the numerous "certain cures" appearing in medical literature, was quite apparent.

In 1816 Henry Ward Beecher who had been a sufferer for many years wrote to Dr. Oliver Wendell Holmes as to his knowledge of a satisfactory remedy. Dr. Holmes could offer no encouragement.

After nearly a hundred years the quest for an efficient therapeutic agent was still being conducted unabated, for in 1903 Dunbar tells us that "I myself have tested all the hay fever remedies on which I could lay my hands within the last ten years but with no one of them did I accomplish a beneficial result." [8]

When certain pollens were recognized as the specific cause of pollen hay fever and pollen asthma, and specific skin tests for specific diagnosis were developed, specific treatment then became possible and promised to supplant the *multitudinous medicaments,* which at the best had only given temporary relief. With specific treatment cases have been greatly benefited and many have been entirely cured. There still remains, however, a large number of patients who, even in the hands of the foremost authorities on the treatment of these diseases have failed to receive any relief whatever from such specific therapy.

The *multitudinous medicaments* are still used by these unrelieved cases. In desperation they seize upon whatever straw of hope floats their way and no group of sufferers is more victimized by the vendors of patented medicines, and *guaranteed* cures which do not cure.

It is for this group that air conditioning offers the most satisfactory relief available. The size of the group can be appreciated from consideration of reports by leading investigators during the past decade. Pursuing a course of treatment aimed at desensitization, several investigators have secured varying degrees of satisfactory results as shown in the accompanying table.[9]

The results show that under the best medical attention perfect results were obtained in only about one-fourth of the total number of cases treated. The

[7] A Survey of the Pollen Flora in Baltimore During 1929, by Paul Ackuarone and Leslie N. Gay, *Journal of Allergy*, Vol. II, 1931, p. 352.

[8] Asthma and Hay Fever in Theory and Practice, by Coca, Walzen and Thommer, p. 739, Chas H. Thomas, 1931.

[9] Results of the Specific Treatment of Late Hay Fever, by Francis M. Rackemann, *J. Immunal*. 2:336, 1931.

Investigator	Percentage Results Consolidated		
	Perfect	Satisfactory	Poor
Cook and Vander Veer	12	52	36
Vander Veer	23	49	28
Walker	25	52	23
Rackemann	10	66	24

remainder, or about three-fourths of all cases can be conceived of as having use for air conditioning at some time during their hay fever season.

The severity of symptoms varies greatly from one patient to another. The same disease that causes only an occasional more or less comical sneeze in one patient may make an invalid of the next, cause great economic loss through disability and expensive measures taken for relief, and if not relieved may cause premature death. Gay has recently pointed out the great economic loss from pollen disease disability and the public health importance of some kind of treatment for the child so affected in order to prevent the development of perennial asthma with secondary upper respiratory infection.[10] Such infections may result in permanent disability, and if unrelieved, threaten the life of the patient as in the case of Van Leuween's first case to be treated by air-conditioned atmosphere.[11]

Scheppegrell in 1924 described a clumsy apparatus for filtering pollen from air supplied to sufferers from pollen disease.[12] Van Leuween in 1925 reported success in having *miasma free* chambers built in the homes of patients who suffer with asthma. The method was so successful that it later became used as an official method of treatment in his clinic.[11] Van Leuween built hermetically sealed rooms, but other workers have found this precaution unnecessary if a slight positive pressure is maintained to prevent the infiltration of air borne pollen. The relief of bronchial asthma by more complete air conditioning was studied by Leopold and Leopold, whereas former reports dealt with simple filtration only.[13] Cohen,[14] Peshkin and Beck [15] treated hay-fever by simple filters designed to remove pollen from the air and also reported success.

More recently Nelson [28] and his coworkers, and also Gay [16] have reported observations upon patients hospitalized and under controlled conditions allowing prolonged observations of patients under medical supervision. The former were successful in giving relief to some 83 per cent of uncomplicated cases of hay fever within 3½ hours.

In evaluating the benefit obtained by the patient it may be well to consider

[10] Economic Results from the Treatment of Pollen Asthma and Hay Fever, by Leslie N. Gay, *Journal of Allergy*, July 1931, p. 456.
[11] Allergic Disease: Diagnosis and Treatment of Bronchial Asthma, by W. Storm Van Leuween, Lippincott, Philadelphia, 1925.
[12] Comfort for Hay Fever Sufferers at Home, by W. Scheppegrell, *Medical Journal and Record*, 120:4, July 2, 1924.
[13] S. S. Leopold and C. S. Leopold: *Journal of American Medical Assn.* 34, p. 731, 1925.
[14] Preliminary Report of Treatment of Hay Fever in Rooms Made Pollen Free by a New Filter, by Milton B. Cohen, *Medical Journal and Record,* March 3, 1926.
[15] A New and Simplified Mechanical Air Filter in Treatment of Hay Fever and Pollen Asthma. by M. Murry Peshkin and Isabel Beck, *Journal of Laboratory and Clinical Medicine,* 15:643, 1925.
[16] Leslie N. Gay: *Journal of American Medical Assn.,* 18, May 6, 1933.

more than merely the temporary relief of symptoms. Gay [10] has pointed out the public health significance of the neglected case of pollen disease and the serious consequence which follows. It is of interest to note in this respect that when Van Leuween constructed his first "miasma free" chamber he was inspired not merely by the sufferings of a case of bronchial asthma, but by the attacks of bronchial pneumonia which all too constantly accompanied attacks of bronchial asthma in a child of 12 years. The encouraging results obtained by Van Leuween in this first case encouraged him to have other patients build such rooms in their homes to which filtered air could be supplied. Thus air conditioning may serve not only to relieve discomfort, but to preserve health in those patients who have not been relieved by other therapeutic measures and who are unable to seek climatic relief.

The favorable reports of relief of hay-fever and asthma by the use of conditioned air have caused air conditioning to be widely seized upon as a cureall, which, of course, it is not. It is a splendidly successful method of giving relief to certain patients who suffer from hay-fever or asthma caused by air borne substances, particularly pollens. Enthusiastic proponents of air conditioning for hay-fever and asthma relief make generous claims which may be difficult to substantiate in many cases. As an agent of relief for hay-fever and asthma, its usefulness is limited, because all cases are not caused by air borne substances.

As early as 1830 Elliotson [17] described symptoms of a woman sensitive to rabbits and in 1864 Salter [18] described his own symptoms upon coming in contact with cats. Seasonal coryza and asthma due to emanations from sandflies has also been described,[19] as have hay-fever symptoms due to cedar,[20] may flies,[21] orris root,[22] dog hair,[23] horse hair and horse dander. Even cuttlefish bones and bed bugs have been reported to have caused attacks of asthma.[24] Hay-fever symptoms have also been reported as caused by the sting of a bee and by attacks of malaria, by animal hair used in furs and by many chemicals used in various industries. The need for specific diagnosis is apparent; and the enthusiast for air conditioning, or simple filtration, should not place himself in the position of guaranteeing relief. Such unfounded optimism can result only in frequent disappointment to the patient and discredit to the principle of air conditioning which can but delay its maximum usefulness as the valuable therapeutic adjunct which it is demonstrating itself to be.

The reports of others have indicated how many hay-fever cases receive relief and how soon such relief is received under hospital conditions both as to the medically supervised selection of patients and their care and observation under treatment. [14,15,16] In the work reported upon in this paper the problem was to determine how much relief could be received by cases suffering with

---

[17] Clinical Lectures, the Lancet, by John Elliotson, 2, p. 370, 1830, as referred to by Coca.[8]

[18] On Asthma; Its Pathology and Treatment, by Henry Hyde Salter, Philadelphia, 1864, Blanchard and Lee.

[19] A Case of Corvza and Asthma due to Sand Flies (Coddis Flies), by S. J. Parlato, *Journal of Allergy,* Vol. I, 1929, p. 35.

[20] Cedar Hay Fever, by J. H. Black, *Journal of Allergy,* Vol. I, p. 71, 1929.

[21] Asthma Due to May Fly, by K. D. Figley, *American Journal of Medical Science,* 178:330, 1929.

[22] Seasonal Hay Fever Not Due to Pollen, by S. M. Feinberg, *American Institute of Medicine,* 3, 1035, 1930.

[23] Perennial Hay Fever; Diagnosis and Treatment, Based on Study of 441 Cases, by Ray M. Balyeat, *Southern Medical Journal,* 22, 492, 1929.

[24] Cooperative Studies in Ragweed Pollen Incidence, by O. C. Durham, *Journal of Allergy,* Vol. I, 1929, p. 12.

hay-fever symptoms who were allowed to visit an air conditioned room at will. In studying the records an attempt has been made to determine the relative relief received by the visitors with respect to the different types of symptoms which the visitors experienced at the moment of entering the air conditioned room.

During the 1933 season approximately 100 hay fever cases visited Dicker Hall. Among these there were 40 cases selected for study. The remainder were excluded either because the records were not complete, or because the records showed that the cases were complicated by other disease. Some persons visited the hall once and left without experiencing relief, and there was no opportunity of following these cases for the completion of satisfactory records. The persons visiting the hall constituted a self-selected group, and came because they wished relief and returned only if they found it. Therefore, the records which have been considered as suitable for study are the records of those who experienced enough relief to make it worth their while to remain in Dicker Hall or to return on succeeding days, and for whom sufficient data were obtained.

The personal records consisted of a brief history of each patient and the type and degree of symptoms present upon entrance and at intervals during the patient's stay in the hall. The various symptoms observed and recorded were as follows:

Symptom Number	Part Affected	Type of Symptom
1	Eyes	Discomfort felt by patient. Itching, burning, sensitiveness to light
2	Eyes	Observed—lachrymation
3	Nose	Discomfort—inside; itching, burning
4	Nose	Congestion, blockage to breathing
5	Nose	Discharge
6	Nose	Discomfort outside; skin of nose, itching. Also other areas of skin as indicated
7	Sneezing	
8		Difficulty of respiration, asthma
9	Mouth	Discomfort, itching, burning
10	Throat	Discomfort, itching, burning
11	Ears, Eustachian tubes	Discomfort, itching, burning
12	Ears	Congestion causing interference with hearing
13	Headache	
14		General depression, fatigue, lassitude

In the discussion which follows, the symptoms have been designated by the corresponding numbers appearing in the first column of the preceding table. Symptoms 8, 11 and 12, however, have been omitted from further discussion and from the remaining tables because these symptoms occurred so seldom as to be insignificant. The subjective symptoms reported by the patient were rated according to severity as follows. Objective symptoms were rated in like proportion.

0 Feels as it does out of hay fever season
1 Minimum, barely observable, not annoying, or if so because of duration rather than degree.

2 Definite and annoying.
3 Painful—Patient is unable to work effectively.
4 Patient is prostrate.

The next table shows the number of visitors, among the 40 selected for study, who complained of a given symptom at some time during the season, and also the percentage of the visitors complaining.

Symptom	Number of Visitors Complaining	Percentage of Visitors Complaining
1	29	72.5
2	19	47.5
3	25	62.4
4	36	90.0
5	32	80.0
6	11	27.5
7	27	67.5
9	6	15.0
10	10	25.0
13	9	22.5
14	21	52.5

This table shows that more visitors complained of nasal blockage (90 per cent) and nasal discharge (80 per cent) than of any other symptom. Next in frequency was irritation of the eyes, followed in order by sneezing, irritation of the mucous membrane of the nose, general depression and fatigue, watering of the eyes, itching of the skin, irritation of the throat, headache, and irritation inside the mouth.

During the 32 days on which records were made, the average number of days attendance in Dicker Hall amounted to only 3.1 days per visitor for those 40 visitors selected for study, making a total of 123 visitor-days, or an average of 3.8 visitors per day. The following table is a consolidation of the daily records and shows the number of patients, or visitors, who visited the air conditioned room on those days when the pollen content of the outside air was as indicated by the arbitrarily-divided ranges of pollen counts as shown in the first column. The table shows also the number of symptoms complained of and the number of symptoms per patient.

Outdoor 24 Hour Pollen Count	Number of Days upon Which Patients Visited Dicker Hall	Number of Symptoms Complained of	Number of Patients Visiting Hall	Number of Symptoms per Patient
0–24	7	49	14	3.5
25–49	7	132	32	4.1
50–99	10	221	53	4.2
100 up	8	105	24	4.4
Total	32	507	123	4.1

The number of symptoms increases from an average of 3.5 on days with pollen counts under 25 grains to 4.4 symptoms for days with pollen counts

of 100 or more per 24 hours. The difference is not as marked as might be expected from the experience of others who have shown how symptoms increase and decrease with the variation in pollen count.[5, 6, 7] However, in these studies by others the measured severity of symptoms was not only for a given time, but also for a whole 24-hour period; whereas, in this study, the severity of a symptom refers to its severity at a given moment, that of entrance into the air conditioned atmosphere.

Durham[24] has shown how much the pollen concentration varies from hour to hour. Such variation makes it possible for an individual to have few symptoms when the pollen concentration is low and to be attacked a short time later the same day when the pollen concentration may have become much higher due to wind or other causes.

The 24-hour count is a measure of the time that the pollen concentration lasts as well as of how great a concentration occurs. Symptoms at any one time, however, depend mostly upon how much pollen is present at the moment. Pollen counts at the moment of occurrence of symptoms would be of interest, but difficult to ascertain by the ordinary method. The experience of having a severe attack following massive exposure to pollen, such as in an open field, is well known.

It may be, however, that, above a given threshold concentration of pollen, relatively low counts such as from 25 to 50 may be found to cause practically the same symptoms as higher counts up until very great concentrations are reached. The data given in the table would seem to support this. If this is true it becomes more important for air conditioning to reduce the pollen concentration as near zero as possible and not to be content with relatively low counts. Relatively low 24-hour counts may allow relief in hay fever resorts over a prolonged period, but for patients visiting an air conditioned room for short intervals the patient's judgment of relief is formed within a short time and the pollen concentration at the moment is important.

The accompanying table shows the frequency of the various symptoms based on the number of patient-visits. Also, the cases have been divided into the numbers which occurred under the second, third and fourth degree of severity. The first degree of severity has been ignored because the relative mildness of the symptoms made it difficult to note any marked change in relief.

Symptom	Degree of Severity			Total Frequency of Occurrence
	2	3	4	
1	61	16	1	78
2	50	3	1	54
3	48	14		62
4	68	43	5	116
5	73	33	1	107
6	14	3		17
7	40	27		67
9	11	1		12
10	17	5		22
13	10	1		11
14	35	8		43

This table shows, as before, that the most frequently occurring symptoms were nasal blockage and nasal discharge, and also that these symptoms were relatively more severe than other symptoms.

In attempting to estimate the relief caused by the air conditioning, certain assumptions must be made which cannot be absolutely substantiated. For example, it cannot be said with certainty that the patient's symptoms would have continued with undiminished severity if he had not sought the relief of conditioned air. Not uncommonly patients gave a history of having had more severe symptoms earlier in the day and that these symptoms had subsided spontaneously before coming to the hall.

Hay fever symptoms notoriously tend to occur spasmodically and attacks tend to terminate spontaneously without apparent cause. Therefore, it cannot be claimed that all of the relief experienced by patients while in the air conditioned atmosphere is caused by the air conditioning.

Also there were numerous circumstances which tended to reduce the degree of relief offered by the air conditioned atmosphere. Patients frequently would leave the hall to visit other portions of the building or even to go outside for short periods and thus delay their relief. When these excursions were for a matter of minutes only, no account was taken of them in the analysis. A most important factor was the possible presence of other unrecognized complicating diseases. Exposure to pollen within the room arose from the fact that it was impractical to prevent carrying pollen on clothing and also because there was some infiltration of unfiltered air. And yet, in spite of these circumstances, the capacity of the air conditioning unit to handle the room air volume several times an hour kept the pollen counts consistently low.

The possible introduction of pollen has been disregarded in this study because no means was provided for testing the theory that patients may have been affected by temporary increases in the amount of pollen in the room. Such increases could conceivably be sufficient to cause symptoms and yet not be sufficient to appreciably affect the pollen count on a 24-hour slide.

In any event, whatever factors tended to influence the symptoms of the patients, only the air conditioning and one other influenced them favorably. All other factors mentioned tend to retard relief. The one other influence was the use of drugs by patients immediately before and during their visit to the hall. Control was impossible; allowance difficult. In the tabulation the use of medicines has been disregarded. Unmedicated cases showed favorable results that tend to justify this policy.

A consolidation of the results obtained by observing the relief secured by the visitors is given in an accompanying tabulation.

This table also shows the length of time patients were in the hall before maximum relief occurred. Symptoms receiving maximum relief under one hour were sneezing, itching of skin, watering of eyes. Other symptoms required over one hour. The time required for relief is important to the patient who has a limited time to spend in an air conditioned atmosphere.

As shown by Fig. 2, the pollen count in Dicker Hall varied on different days from 0 to 14 grains. An attempt was made to correlate the relief obtained in

Symptom	Per Cent Relief	Time in Minutes Required for Relief
1	65	56
2	54	51
3	59	51
4	40	76
5	62	61
6	66	46
7	91	30
9	85	69
10	50	100
13	47	108
14	45	83

the hall with the inside pollen count. A summary of the results is given by the accompanying data:

Pollen Count Inside Hall	Visits Giving Some Degree of Relief	Visits Giving No Relief	Percentage of Visits Giving No Relief
1–4	377	35	8.49
5–9	117	14	10.68
10–14	38	7	15.55
Total	532	56	9.52

The data show that there was a consistent increase in unrelieved cases corresponding with the increase in the pollen counts. While not convincing these data strongly suggest that even small amounts of pollen interfere with relief in an air conditioned atmosphere. If this is true it makes it more desirable to reduce the amount of pollen introduced into the room on clothing and by infiltration.

Durham[25] has reported that pollen counts over a large area have chronological peaks corresponding to wind and storm in the area. In the present study it has been shown that rain causes an increase in the pollen count, especially when followed by sunshine. Observations over shorter periods than 24 hours are needed to determine the variations before, during and following rain. During this season a protracted rainy period was accompanied by low pollen counts. Counts seem to be increased by the wind before and the sunshine following, but decreased by the rain itself.

Moskow and Spain[26] have noted that the patient's discomfort from hay-fever is decidedly increased upon days of very high temperature and low humidity. Their studies did not include estimates of pollen concentration.

In order to study the relationship between the relief of symptoms and the conditions within Dicker Hall an analysis was made of the relief obtained in relation to various conditions of temperature, relative humidity and pollen

[25] The Contribution of Air Analysis to the Study of Allergy, by O. C. Durham, *Journal of Laboratory and Clinical Medicine*, 13, 967, 1928.
[26] Ragweed Hay Fever, by Harry Moskow and W. E. Spain, *Journal of Allergy*, Vol. I, 1930, p. 414.

counts for symptom No. 4 which most frequently failed to be relieved. The analysis shows that failure of relief of the symptom occurred practically throughout the entire range of temperatures, relative humidities and pollen counts for which relief was experienced. If the data justify any provisional conclusion it is that effective temperatures below 70 deg are less favorable for the relief of nasal congestions than the higher temperatures. A summary of the relevant data is shown in Table 3.

Considering the relative humidity alone, the results show that for a relative humidity of from 65 per cent to 69 per cent the percentage of visits giving

TABLE 3

Number of Pollen Grains per Sq. Cm. per 24 Hours	Effective Temperature F.	Degree of Relief Obtained							
		No Relief or Made Worse				Satisfactory Relief			
		Per Cent Relative Humidity				Per Cent Relative Humidity			
		65 to 69	70 to 74	75 to 79	80 to 84	65 to 69	70 to 74	75 to 79	80 to 84
1 to 4	65 to 69	1	0	1	0	1	4	2	0
	70 to 74	3	6	5	2	12	24	12	7
5 to 9	65 to 69	0	1	1	0	0	3	0	0
	70 to 74	0	0	0	0	6	10	6	0
10 to 14	65 to 69	0	0	1	1	0	0	2	1
	70 to 74	0	1	0	0	0	1	0	0

relief to nasal blockage amounted to 88 per cent. The relief obtained became less as the humidity increased so that for humidities above 80 per cent the percentage of visits giving relief amounted to only 70 per cent.

If all symptoms were equally unpleasant it would be desirable to plan relief for those which occurred most frequently. Actually No. 4, nasal blockage, was one of the most distressing symptoms, occurred most frequently and was least often relieved.

Patients frequently reported that on the day following their visit to the hall that their symptoms were less severe than usual. Cause for such milder symptoms is difficult to determine. Symptoms have been demonstrated to vary widely from day to day, but to follow the rise and fall of the pollen count. Blackley [4] was able to predict from his own symptoms each day the approximate amount of pollen he would find on his slides. The Rev. Henry Ward Beecher noticed over 50 years ago that his symptoms seemed to be worse on alternate days. The variation in the daily pollen counts is the now accepted explanation of this irregularity of symptoms. Patients, however, experienced relief after a previous day's visit when the pollen count was too high to account for the relief. This suggests a prolonged benefit due to exposure to the air conditioned

atmosphere, but there was no opportunity to study this aspect of the problem in detail.

Duration of relief could well be studied from the standpoint of the time of day the patient left the shelter of the hall and the outdoor pollen count at that time. Such pollen counts for special times were not available for this study. Durham [26] found that such counts made at hourly intervals through the day showed the count to be above the mean concentration from about 9 a. m. to 9 p. m. and below from 9 p. m. to 9 a. m. When patients visited on consecutive days and when data could be obtained with reasonable accuracy a record was made of which symptoms returned after the patient left the hall and what time lapsed before the return of such symptoms. It was observed that patients who left Dicker Hall in the evening did experience relief for a longer time than those who left during the day. This is what one would expect from Durham's findings. It must be noted, however, that the patient who left in the evening was more likely to go indoors soon after leaving.

It was noted that when symptoms started shortly before the patient entered the air conditioned atmosphere or when patients left the comparative shelter of an indoor atmosphere to come to the hall, symptoms occasionally became worse for a short time after the patient entered. If the patient remained as long as one hour symptoms usually subsided but occasionally patients experienced increased symptoms and left the hall in discouragement without allowing sufficient time for a satisfactory test.

As the result of experiments of blowing dry pollen into one nostril of hay fever patients Efron and Penfound reported that most cases react within a few minutes after the test is performed, but symptoms in some cases were observed to be delayed as long as 14 hours after this test.[27] Occasional increased discomfort experienced after entering the air conditioned atmosphere may be partially accounted for by a delayed action of the pollen received while exposed in the open air. Nasal blockage especially was the cause of increased discomfort not only to patients upon entering the hall, but also after prolonged visits within the hall. It was noted that this symptom tended to reoccur in the same individuals and it would therefore seem to be dependent upon some peculiarity of the individual patient. The fact that other symptoms could be relieved in these cases showed that the air conditioning was effective. The persistence and even aggravation of nasal blockage in occasional patients impressed the observers with the need for a thorough medical study of each patient and a conservative policy with respect to assuring a patient that air conditioning will relieve all of his symptoms.

Even with what seemed to be favorable cases, air conditioning may not relieve all symptoms because of some unsuspected complicating disease in addition to the hay fever. Occasionally patients were recognized to be suffering from some concurrent infection that interfered with their hay fever symptoms. Such conditions as head colds, nasal polyps, hordeolum, peri-tonsillar abscess, etc., cause the persistence of symptoms which otherwise would probably be relieved if due to uncomplicated hay fever. When such conditions were recognized the records were not used in this study. Without doubt many such complicating conditions were overlooked because it was not possible to obtain a

[27] A Nasal Test with Dry Pollens in the Diagnosis of Seasonal Hay Fever, by B. G. Efron and W. T. Penfound, *Journal of Allergy,* Vol. I, 1931, p. 43.

complete medical diagnosis. Such undiagnosed cases included in this study tend to reduce the measure of relief obtained below that which uncomplicated cases might expect.

One such case is of special interest because it suggests the possible use of an air conditioned atmosphere as a diagnostic aid to the physician contemplating nasal surgery, and a protection to the patient following such surgery. A patient with hay fever presented himself with symptoms of nasal blockage. The physician wished to avoid surgery during the hay fever season for two reasons. First, because of the difficulty of determining how much of the disease was caused by a condition to be benefited by surgery and how much was caused by the patient's hay fever; and second, because nasal surgery during the hay fever season has been reported as a cause of hay fever and asthma. When the patient experienced relief from all hay fever symptoms except nasal blockage in an air conditioned atmosphere, the condition was considered a surgical one and the surgeon operated with good results. Following the nasal operation the patient spent as much time as possible in the pollen free atmosphere of the air conditioned room, a procedure which might well be made routine following nasal and throat surgery during the pollen season.

A second case is of interest because the patient suffered with little relief on the first and second day of her visit and was much discouraged with air conditioning until it was discovered on the third day that she was suffering from an abscess in her throat. After suitable treatment of the throat, her hay fever improved.

It has been noted that subjective symptoms are relieved in an air conditioned atmosphere more quickly than are objective symptoms.[28] This has been explained as due to the persistence of the tissue injury for a limited time after the cause of the injury (the pollen) has been removed. The same report called attention to the observation that at the onset of the hay fever season the offending pollen was found to be present in the air in appreciable amounts for days before patients complained of symptoms. Likewise the pollen disappeared from the atmosphere for a period as long as two weeks before patients were relieved of the discomfort caused by the prolonged pollen irritation of the mucous membranes. It has likewise been noted by patients that if they delay going to their hay fever resort until symptoms have started, that while they obtain early relief a certain amount of residual discomfort persists for a period of a week or more. One should therefore not expect air conditioning to immediately relieve these symptoms due to changes in tissue when these same symptoms are not immediately relieved by hay fever resorts nor by the absence of pollen at the end of the season.

Thus a number of patients failed to experience complete relief even after prolonged visits. If their relief were equal to that experienced during the days immediately following the close of the season it would seem logical to give greater credit to air conditioning than the simple record would indicate. These patients who are not entirely relieved may be those who have developed tissue changes which in turn cause secondary symptoms, and indeed from the history of a few cases this seems probable. Even though not complete this relief is often marked.

---

[28] Hay Fever and Pollen Asthma; The Effect of Air Filtration, by Tell Nelson, B. Z. Rappaport and William H. Welkers, *Journal of American Medical Assn.*, 18:1385, 1933.

Another cause of only partial relief of symptoms may be that some cases have hay fever out of the pollen season, but find that symptoms are made worse by pollen. The removal of the pollen in such cases would be expected to give only partial relief and again the need for a medical history and diagnosis and a conservative policy is indicated. Here also, however, the relief obtained is definite and valuable.

Some cases seeking relief from symptoms of hay fever, are actually made worse by the air conditioned atmosphere. This observation has been noted before [18] and again impresses one with the fact that air conditioning should not be offered indiscriminately to all types of patients, suffering with what appears to be hay fever or asthma. Elderly persons and hay fever patients who also suffer with asthma may be made worse by the refrigerated atmosphere and for these cases simple air filtration may be advised.

Opportunity was offered occasionally for patients to compare the relief obtained in a private residence using a single air filter with the relief obtained in the air conditioned room. In one such case the patient experienced greater relief in the air conditioned room even though the pollen count made at the residence was lower than that in the cooler air conditioned room. The pollen counts were 1 and 0 for the residence and 2 and 3 for the air conditioned room. The temperature in the air conditioned room however was some degrees lower than that in the residence and this seemed to make the difference to the patient.

Two patients upon their first visit entered the hall while the spray water was turned off and the air was passing only through the filters. After 45 min they had experienced no relief and expressed disappointment at this time. Without notice to them the water spray was put into operation. Both patients remarked upon the decrease in their symptoms immediately after the spray water was started. They were near the machine and in position to notice the change in the temperature of the delivered air.

It was impossible to determine the comparative relief experienced by the same patient in the refrigerated atmosphere of the Hall and in a room adjacent to Dicker Hall equipped with a simple filter because patients rebelled against remaining in the warm room when the cooler room was available.

Efron and Penfound have reported that when dry pollen was introduced experimentally into the nostrils of patients most cases reacted within a few minutes.[27] Unfortunately for the patient seeking and finding relief in an air conditioned room away from his home, this experimental reaction has its counterpart in nature and symptoms frequently return with distressing promptness after the patient leaves the air conditioned atmosphere. Fränkkel [29] has described a portable filter mask which might prove useful to patients going out of doors for short periods.

It was noted that the time required for the return of symptoms varied not only from patient to patient on the same day, but also for the same patient. Other than the patient's tolerance, the pollen concentration was probably the most specific factor involved.

Many hay fever sufferers have noted a comparative mildness of symptoms as long as they remained within their homes. They have noticed that by

[29] Experiments with Filter Apparatus with Allergic Patients, by Ernest Fränkkel and Else Levy, *Klin Wchnscher* 7:2292, 1928.

reducing the ventilation of their homes they can reduce the severity and the number of their symptoms. If such patients have severe attacks upon going out of doors they plan to remain in doors as much as possible. This may be the very type of case that is most relieved by a visit to the air conditioned room, but because of the attack suffered while on the way back home the patient prefers to remain at home for days at a time rather than to risk the more severe attacks which follow the greater but temporary relief experienced in the air conditioned room.

SUMMARY

The results of this work may be summarized by the following statements which appear to be substantiated by the personal records of the visitors to Dicker Hall:

(1) All of the 40 visitors selected for study received some degree of relief in the air conditioned room. Satisfactory relief was obtained upon about 90 per cent of all visits made by these patients.

(2) Records of other visitors were unsuitable for study because in some cases symptoms were of a degree too mild for the observation of relief, and because in other cases in which relief was obtained the patient either could not be kept under observation or other pertinent data could not be obtained.

(3) Some cases received relief while in the room but failed to visit the room often enough to allow for sufficient study. Two principal reasons given for failure to visit the room were:

(a) lived too far away, and hence not convenient to come; and
(b) attacks appeared on the way to and from the hall, and hence preferred to remain at home.

(4) The various common symptoms usually experienced by hay-fever sufferers received different degrees of relief.

(a) Sneezing and irritation of mucous membrane received the greatest amount of relief.
(b) Irritation of the eyes, facial irritation and nasal discharge received a moderate amount of relief.
(c) Nasal blockage, headache, and general depression or fatigue received the least amount of relief.

(5) The data do not show conclusively how much relief is due to filtration only and how much is due to the more comfortable temperature and humidity which prevailed within the air conditioned room as compared with less comfortable conditions in other rooms. The fact that the patients *preferred* the air conditioned room would merely confirm the judgment of normal persons.

(6) There was occasionally an exceptional case that was made more uncomfortable in the cooler atmosphere of the air conditioned room.

(7) The observers were impressed with the need in all cases of medical diagnosis and sensitivity tests to determine the specific cause of symptoms.

DISCUSSION

H. C. MURPHY (WRITTEN): In the authors' summary the statement is made that "The data do not show conclusively how much relief is due to air filtration only and how much is due to the more comfortable temperature and humidity which prevailed in the air conditioned room."

I would like to call attention to two very exhaustive recent studies which are apparently rather conclusive on this point.

Dr. L. N. Gay in the summary of his extensive tests at Johns Hopkins as reported in the May 6, 1933 issue of the *Journal of the American Medical Association* has the following: "The relief of patients apparently depends not only on the cooling of the atmosphere but also and chiefly on the fact that the atmosphere is cleaned of the offending agents."

In the same issue of the *Journal*, Drs. Nelson, Rappaport, and Welker state as

the results of an investigation covering two years in carefully controlled wards at the University of Illinois Hospital at Chicago: "What is the effect of cooled filtered air on the time required for the relief and the degree of relief? Of the entire group studied in both rooms, 72.8 per cent were relieved of their symptoms in three hours or less. In the cooled ward, the proportion was only 61.1 per cent, while in the uncooled ward it was 76.9 per cent. Several of the patients were shifted from one ward to another during their stay in the hospital. All stated that the degree of relief in the uncooled ward was greater and the time necessary for this relief shorter than in the cooled ward."

Under "Conclusions" they state, "cooling of filtered air may aggravate asthma because of an increase in relative humidity."

These opinions agree very well with our previous knowledge of these disorders.

In his recent work on Allergic Diseases, Dr. W. Storm Van Leuween of Leiden, Holland, states: "Our first publication on a miasmfree room was made in April, 1924. (*Proc. Royal Sec. of Medic.*, Vol. XVII.) The principles laid down in that publication have been worked out so completely that we have opened a clinic for allergic diseases in Leiden, based entirely on the principle of pure air treatment in miasmproof chambers. The results of this treatment have by far exceeded our expectations." He further states that it is his belief that asthmatics of the *climate* type, that is, asthmatics subject to an unknown group of allergens occurring in the air, will be free from attacks whenever they are in a room which is hermetically closed, but ventilated continuously with air filtered so as to be free of miasms. He discusses the effectiveness of this treatment and states that when, without the patient's knowledge, the filtration of the air was defective, attacks re-occurred: "One day the motor which worked the ventilation of the room was defective. Twenty-four hours later the child had asthmatic attacks every two hours. After the repair of the motor she returned to her original condition again in three days."

From the time of our earliest knowledge of hay-fever and asthma, they were recognized as *diseases of climate* and relief was sought by travel. In this manner some localities were developed as hay-fever resorts and sufferers with sufficient means made it a practice to spend the hay-fever season at these favored spots. This, of course, is an expensive and often inconvenient procedure beyond the reach of many persons, for which perhaps hay-fever and asthma are frequently referred to as a *rich man's disease*.

Possibly the most famous resort of this type is Bethlehem, N. H. in the White Mountains. It owes its efficiency to the altitude and abundance of pine trees which produce shade and cover the ground with pine needles preventing the growth of grass and weeds.

The success of other famous resorts in this respect, such as Mackinac Island in Northern Michigan, also depends upon the absence of pollenating weeds and grasses.

H. B. JONES: This year Dr. Rappaport worked with direct absorption of the moisture, with the air carrying lower humidities than a year ago, and found that he gave almost complete and instantaneous relief for hay-fever within 15 min. Sometimes two to three days were required to clear up asthma cases, but he found that the asthma would sometimes reappear and those reappearances frequently were coincidental with thunderstorms. Whether it is merely coincidence or whether there is some reason for it, he doesn't know, but it is interesting and presents a subject for further study. After being completely free from it, a number of the asthma patients had a reoccurrence of their trouble, just before and during the first part of a thunderstorm.

CHARLES DAVIES: The studies that have been made in this paper, in which an air conditioning unit was used for hay-fever relief, show that there is very little difference in the result in hay-fever relief obtained. In 1930 and 1931 studies were made and a paper was written by Isabelle Beck and Dr. Peshkin of Mt. Sinai Hospital

and the percentage of relief obtained by hay-fever sufferers was practically the same. It seems to me that the only advantage of an air conditioning unit is simply that of the extra comfort that might be obtained in reducing the temperature.

It was found, however, that if the temperature was reduced too much, if there was too much differential of temperature between the outside and inside, the sufferer would notice that symptoms increased, especially in the case of asthma.

There are many hay-fever sufferers, who before the advent of the small unit for the home, would spend many hours in movie houses with the object of obtaining relief. One or two physicians who had hay fever and knew that pollen caused it, knew definitely that they could be relieved by going to movie houses, but they found that their symptoms were considerably aggravated when they left, due to the differential in temperature.

It seems, therefore, that air filtration is the thing for hay-fever relief and that air conditioning can be used, but I think it should be used in such a way as to produce only a slight reduction in the humidity and temperature.

In this paper the authors did not mention a very important fact and that is that there are a very large number of hay-fever sufferers who, if they spend sufficient hours in the pollen-free air, have a noticeable carry-over of relief, that is to say, they fare very comfortably during the night when they are in an air filtered atmosphere and when they leave the room, contrary to the general idea they do not get their symptoms immediately, but a very large number of them can carry over relief during quite a long period of time. This is due to a certain cumulative effect of pollen on the system.

I do not believe that the investigation made by having patients come in during the daytime and spend a certain number of hours allowed for very exact control. As people who have hay-fever can hardly be expected to install large equipment in their homes, I think if the manufacturers of units would send out questionnaires to all users of small air filter units and ask them to give a complete description of the relief they obtained they would get much better information because they would have the data in sequence.

J. H. WALKER: I was given to understand that there were different types of asthma; that some of those types are aggravated by air conditioning, by the condition of the air, while the other types are aggravated by conditions in the blood or nervous system. I was wondering if those different types of asthma had been analyzed or defined sufficiently so that in considering the treatment of asthma with air conditioning equipment we know which type of asthma we are treating. Possibly there is someone here who can tell us more about that.

MR. DAVIES: I have had a lot of experience with this and believe that there are many different types of asthma and hay-fever. Although hay-fever is caused by pollen, that is the hay-fever we know, the hay-fever of August or September or the rose fever of May, June and July, the individual sensitivity is sometimes increased towards pollen by outside factors such as foods, furs, feathers, dusts, etc. There are people who during the hay-fever season become extremely sensitive to pollen in the presence of certain foods. I know of a man who couldn't eat a strawberry during the summertime because this would cause an attack of sneezing and itching eyes, even if he was in a pollen-free room.

The true value of pollen-free air in the treatment of asthma is not yet definitely known.

MR. JONES: In the work that Dr. Rappaport did in the Illinois University Medical School at Chicago, he endeavored to eliminate such asthma cases as did not show definite traces toward the pollen. In other words, if a patient had asthma the year 'round he didn't try to treat it at that clinic. He was trying to study the allergic diseases and he eliminated other patients. Undoubtedly there are an infinite number of causes of asthma.

No. 983

CORROSION STUDIES IN STEAM HEATING SYSTEMS

By R. R. SEEBER,* F. A. ROHRMAN ** AND G. E. SMEDBERG † (*NON-MEMBERS*)
HOUGHTON, MICH.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers and conducted at the Michigan College of Mining and Technology

THIS paper presents the progress made to date on the investigation of the relationship of oxygen, carbon dioxide, pH value and method of operation on the corrosive properties of condensate in steam heating systems. Previous attempts have been made to determine these factors in a regular plant, but it was deemed necessary to devise a laboratory set-up in which these variable factors could be controlled to a certain degree and be determined accurately.

With the cooperation of the Technical Advisory Committee on Corrosion of this Society and of the Corrosion Committee of the *National District Heating Association,* both of which are under the chairmanship of J. H. Walker, a set-up of this type was designed and erected at the Michigan College of Mining and Technology. It is an elementary heating system, containing one radiator with supply and return piping and with means for measuring the corrosiveness of the conditions at different points when the system is operated in various ways. The variable conditions which will first be studied are:

1. Constant pressure on radiator, vary vacuum on return.
2. Various pressures throughout system. (20 in.-2lb gage.)

The description of the apparatus is as follows. The steam is generated in a 66-gal vertical boiler, by means of two immersion type electric heaters having variable heating ranges. (See Fig. 1.) The pressure or vacuum in the system is controlled by a thermostat located on top of the boiler and operated through a relay for opening or closing the circuit to the heaters as the pressure varies, and by means of a vacuum control on the top of the receiver for opening and closing the circuit of the vacuum pump. From the boiler, the steam flows through two *corrosion loops,* a 40-sq ft radiator, and two more *corrosion loops,* and then to a 16-gal receiver. When the receiver is three-quarters full, a float

* Professor of Mechanical Engineering, Michigan College of Mining and Technology.
** Assistant Professor of Chemistry, Michigan College of Mining and Technology.
† Research Mechanical Engineer, Michigan College of Mining and Technology.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by R. R. Seeber.

switch in the receiver turns on a water pump and the condensate is forced back to the boiler. When the water level falls to the half-mark, the switch turns off the pump. This pump is by-passed and used only when the return line is operating under a vacuum. Packless valves are employed above the water line in order to eliminate infiltration of atmospheric gases. Fig. 2 shows a complete plan of the apparatus.

Oxygen, carbon dioxide, air or other gases can be bled into the system through special fittings located on the water feeder line. In this way a variation in the amount of oxygen and carbon dioxide can be obtained.

The corrosion testers used in each of the four loops are the type devised and recommended by the *National District Heating Association.* Fig. 3 shows the method of installation of these testers and also the type which was designed at

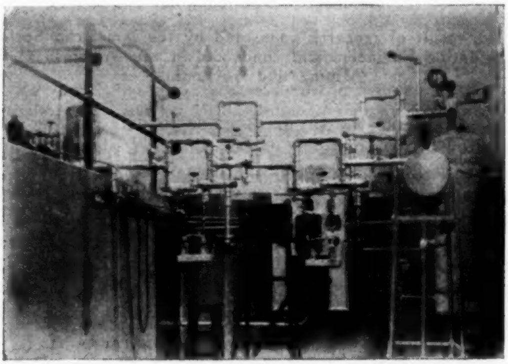


FIG. 1. APPARATUS USED TO GENERATE STEAM

the Michigan College of Mining and Technology, which is to be used later. The *N. D. H. A.* testers consist of three helical coils of 0.05 in. diameter Bessemer steel wire suspended on an insulating frame. The M. C. M. & T. tester consists of a pipe section which fits into the union of the loop. The testers are first weighed and then placed into the apparatus for the required period of time, after which they are removed and cleaned of their products of corrosion by boiling in a mixture of sodium hydroxide and powdered zinc. They are then washed, dried in acetone, and re-weighed. From the loss the corrosion rate is calculated from the following formula:[1]

$$R = \frac{A}{B \times C}$$

where

R = the corrosion rate as average penetration in inches per year.
A = the loss in weight in pounds per square inch.
B = weight of metal in pounds per cubic inch.
C = duration of test in years.

[1] Formulas by *N. D. H. A.*

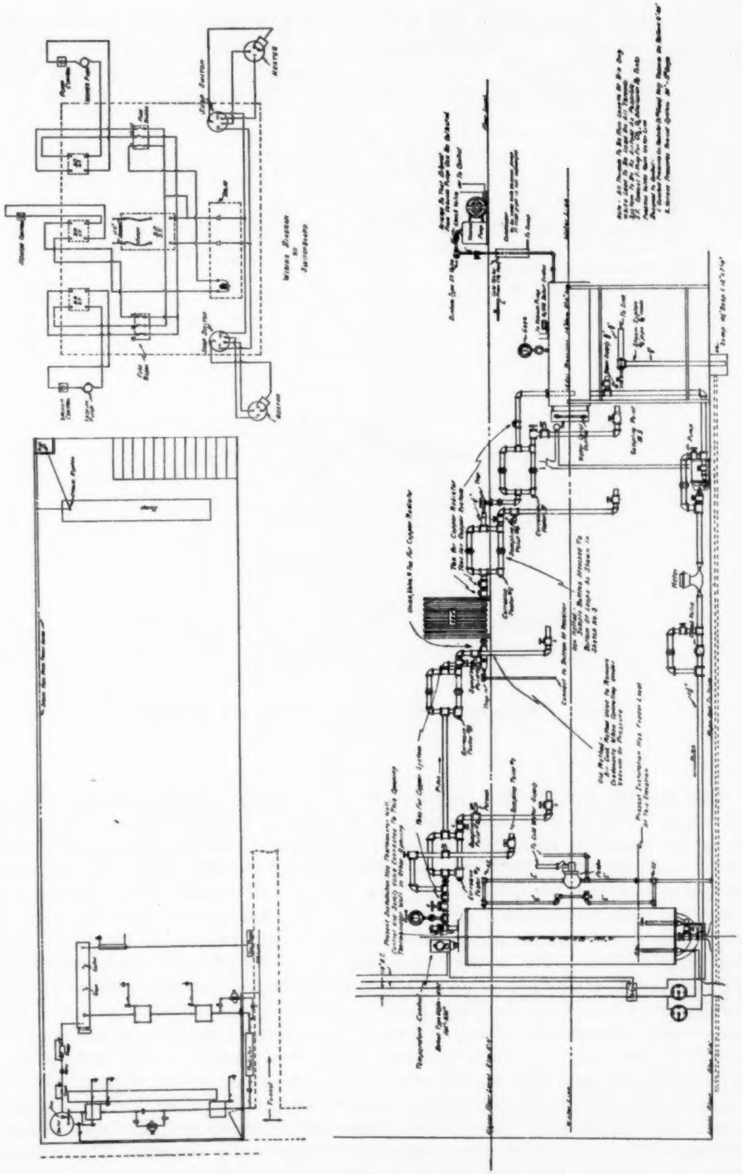


FIG. 2. COMPLETE DIAGRAM OF APPARATUS

However, if the corrosion rate is very high, a formula which considers the correction necessary for the reduction in area exposed to corrosion as the process proceeds is used. This correction must be taken into account with corrosion rates greater than 0.02 in. per year. The formula is as follows:

$$R = \frac{D}{2T}\left[\frac{W_1 - W_2}{W_1 + \sqrt{W_1 W_2}}\right]$$

in which

R = corrosion rate, inches penetration per year.
D = original diameter of wire, inches (0.05).
T = time of exposure, years.
W_1 = initial weight of wire.
W_2 = final weight of wire.

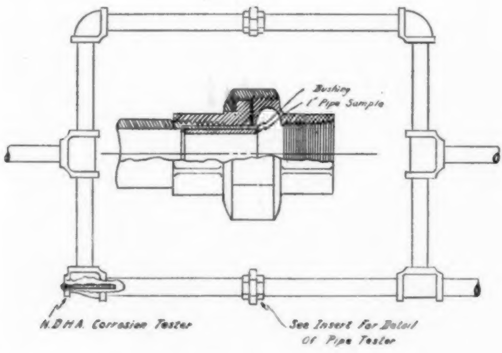


FIG. 3. METHOD OF INSTALLING TESTERS IN CORROSION APPARATUS

The difficult part of this investigation is to obtain a representative sample of water from each of the respective loops, and after obtaining it to keep it in this condition until the analysis is completed. The first method used was to fill each of the vertical pipe sections attached to each loop with nitrogen by means of the petcock, leaving the lower valve open until all the air contained was forced out. (See Fig. 4.) The lower valve and the petcock were then closed and the upper valve opened so that the condensate flowed into the pipe section. The upper valve was then closed and a nitrogen-filled Erlenmeyer flask was placed under the lower valve, the petcock opened, and the condensate forced into the flask to overflowing by the nitrogen pressure and then tightly stoppered. This method was considered to have a tendency to possibly force some of the gases out of solution because of the high partial pressure of the gas, and the nitrogen would gain access to the system. In the second method used, the condensate samples were removed by forcing out any water left in the pipe section by means of nitrogen gas and the section evacuated by means of a water siphon attached to the petcock. The petcock was then closed and the upper valve opened, allowing the condensate to fill the

pipe section. The nitrogen-filled bottle was then placed under the lower valve and the upper and lower valves opened so that the pressure of the system forced the condensate sample out. This method could only be used under pressure conditions and also had some danger of the oxygen leaving the solution as it entered the evacuated pipe section. After these two attempts, and at the suggestion of Chairman J. H. Walker, it was deemed advisable to place the bottles for the collection of the condensate in the line and remove them when they were to be analyzed for oxygen, carbon dioxide, and pH value. (See Fig. 5.) This figure shows the method of attaching the bottles to the *corrosion loops* and also a description of the method of operation. For convenience, the method of operation is repeated here:

Bottles H and I are attached to the loops as shown. Stopcocks D, E, F, and G are opened. Hose from water suction is attached to petcock J. Bottle train is

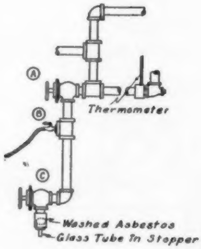


FIG. 4. APPARATUS USED TO OBTAIN SAMPLES OF CONDENSATE FOR ANALYSIS

Method No. 1—With valve *A* closed and *C* open the pipe section is filled with nitrogen gas through petcock *B*. Valve *C* closed and also petcock *B*. Valve *A* opened and condensate flows into pipe section. Valve *A* closed. Nitrogen-filled bottle placed under valve *C*, petcock *B* opened, also *C*. Nitrogen gas forces condensate into bottle. Bottle removed and stoppered.

Method No. 2—Valve *A* closed, *C* open. Condensate in loop from last test forced out by nitrogen gas from petcock *B*. Valve *C* and petcock *B* closed. Hose from water suction pump attached to *B* and pipe section evacuated. Petcock *B* closed and valve *A* opened. Condensate flows into pipe section. Nitrogen-filled bottle placed under valve *C* and *C* opened. Pressure of system forces condensate into bottle. When filled, bottle is removed and stoppered.

evacuated and petcock J closed. Valve A is closed or partially closed, depending on which operated better during the test. Valves C and B then opened slowly, permitting the glass to rise in temperature slowly. Bottles are filled with the condensate. Valve A then opened. Valves and stopcocks B, C, D, E, F, and G are then closed. Bottle group then removed by disconnecting of points 1 and 2. Bottle group then removed through trap door in bottom of bottle stand. The two bottles are separated from each other at point K. They are then placed in cooling vat and cooled to 20 C. The $MgSO_4$, KI, and H_2SO_4 are added to the 250 cubic centimeters bottle (H) by means of the stopcock D, opening E slightly if necessary. Stopper is then removed from bottle H and quantity reduced to 200 cubic centimeters and the starch solution added. Then titrated with $Na_2S_2O_3$ to end point. Sufficient condensate is withdrawn from bottle I by opening stopcocks G and F so that 100 cubic centi-

meters remain. This amount withdrawn is analyzed immediately for pH value. To the 100 cubic centimeters remaining in the bottle I is added 25 cubic centimeters $Ba(OH)_2$ solution to stopcock G. Stopper then removed and phenolphthalein solution added. Titrate to end point with HCl acid.

As explained on this diagram the analysis is carried on under the most exacting conditions. All the reagents are added through the stopcocks and the

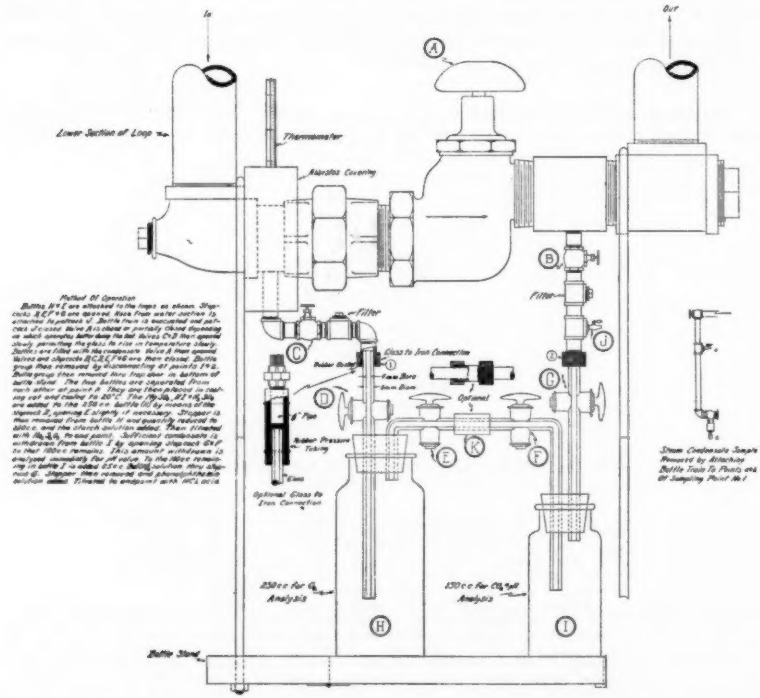


FIG. 5. METHOD OF ATTACHING BOTTLES TO CORROSION LOOPS

possibility of air contamination is very small. In all previous tests the analysis was determined every day during the test. During the present test, the analysis has been run every other day and the condensate permitted to flow through the bottles for a period of three hours before removal. This is done to prevent any discrepancy due to the bottles being filled with air when installed in the system. After the bottles are removed they are cooled to 20 C before analysis is begun.

Since it was necessary to use rubber stoppers in the bottles, the question arose whether the use of this type of stopper would cause any contamination.

A series of tests were conducted which showed that the possibility from this source was not a factor.

Up to the present time we have completed twelve tests of one week duration each. The present test is of three weeks duration. In the first eleven tests the condensate was withdrawn by methods one and two, that is, the first five tests by method one and the remainder by method two. In test number twelve and the present test, the method using the bottles attached to the loops was used. The first tests, although of short duration, helped to point out the discrepancies in the method of withdrawal of the condensate samples. Curves were plotted which showed that the oxygen concentrations obtained were not constant for constant corrosion rates.

This method of analysis with the corrections necessary for the gases in the reagents used will, in our opinion, give accurate, representative results, and, having developed test methods and technique of operation the progress of the work will now go forward with more rapidity.

The main purpose of this progress report is to invite criticism and the authors will appreciate any helpful suggestions relating to this study.

DISCUSSION

J. H. WALKER: I want to tell you of some of the previous work on corrosion, so that you may have a better understanding of the significance of Professor Seeber's paper. It is generally agreed that corrosion in steam heating systems is usually

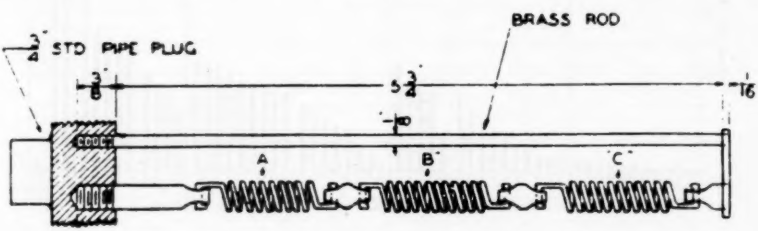


FIG. A. CORROSION TESTER

due to the presence of oxygen and carbon dioxide acting either separately or in combination, but we have very little information regarding the quantities of those two dissolved gases which may cause corrosion. Furthermore, there seems to be some difference in the corrosion in different kinds of heating systems and in different parts of the same heating system.

The *National District Heating Association,* which has been studying this problem for a number of years, has developed a corrosion tester for measuring the degree of corrosion under different conditions. This corrosion tester shown in Fig. A consists of three coils of Bessemer steel wire, which has been manufactured under specific conditions and has a known history. These three coils of wire are supported on a frame and separated from the frame by insulating connections. The frame is mounted on a ¾-in. pipe plug and can be introduced into any part of a

heating system. The usual method of installation is to install the tester in a horizontal pipe such as a return line and cause the condensation to flow over the coils. The weight of the coils is taken before installation and the tester is exposed for about 4 weeks. The coils are removed, the corrosion products are cleaned from the wire and they are reweighed. The loss in weight is calculated as inches of penetration per year and multiplied by one thousand to bring it up above the decimal point. That figure, then, is an index of the inherent corrosiveness of the conditions to which the tester has been subjected. We haven't yet tried to translate these figures into terms of life of pipe or anything of the sort. They simply afford a relative comparison between the corrosiveness of different sets of conditions.

Fig. B shows the results of some of the testers that have been installed in various parts of the country. Altogether we have put out approximately 75 of them and we are gradually accumulating enough data so that, knowing the results ob-

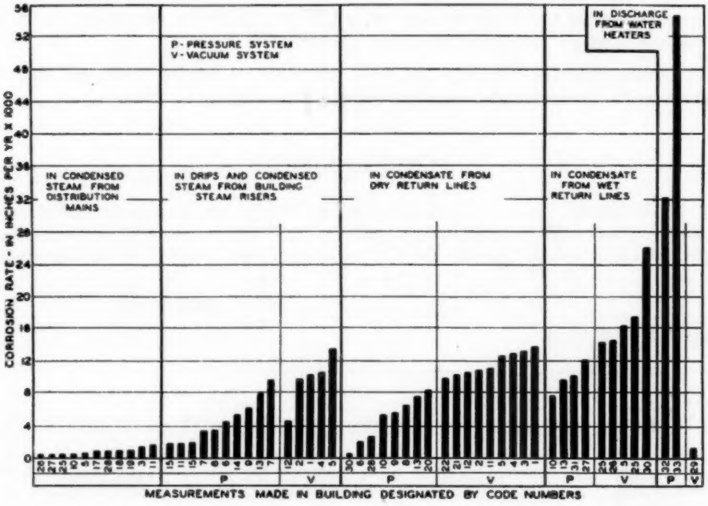


FIG. B. CORROSION RATE MEASURED BY TESTERS

tained with a tester, we can now approximately predict the amount of corrosion that might be expected; that is, whether corrosion will be active or negligible. The height of these lines (in Fig. B) shows the corrosion rate measured by the testers installed under different conditions. The first group were installed in places where they were subjected to the condensed steam from the distribution lines of district heating companies. It is a well-known fact that very few district heating companies have ever experienced internal corrosion of their piping systems, even though those pipes have been in the ground for as long as 30 or 40 years. The trouble seems to be confined almost entirely to individual buildings and it is by no means limited to buildings served from district heating plants.

The other groups show the corrosion found in various other locations and it is quite apparent that there is a vast difference in the amount of corrosion experienced in different types of heating systems and in different parts of any one heating system.

When Professor Seeber told us that he was anxious to undertake some work on corrosion and asked for a suggestion, we recommended the building in the laboratory of a model heating system in which two questions could be studied: namely, the effect of different amounts of oxygen and carbon dioxide and the effect of different methods of operation on this model heating system. It was felt that the comparison between different buildings having different types of heating systems was not very fair because there are other variables that could not be controlled. The *N.D.H.A.* fair because there are other variables that could not be controlled. The *National District Heating Association* corrosion tester is being used by Professor Seeber for his corrosion measurements.

Professor Seeber has prepared this paper which consists of a description of the apparatus and he invites your suggestions and criticisms. I want you to understand that the conducting of work of this sort requires a very high degree of skill. The chemist has to differentiate between a very few parts per million of the dissolved gases. The *National District Heating Association* has published a book of about 30 pages devoted solely to suggestions and instructions for extracting samples of condensate and analyzing them. It is just that difficult a problem.

A. A. Adler: The data here presented seem to show that corrosion is more severe in vacuum than in gravity heating systems. This appears contrary to the view held by some operators of power boilers. In these cases corrosion seems to take place entirely below the water line and about in proportion to the amount of oxygen and carbon dioxide present. According to Henry's law the solubility of gases is directly proportional to the superimposed pressure. It also decreases with increase of temperature. With this in mind corrosion should be less in vacuum systems. However, the facts so far determined by the authors seem to show that the opposite is true. My own observations, admittedly not very extensive, show that very little if any corrosion takes place in the steam space of a boiler.

Accordingly I offer the following suggestions to the authors for their future research: (1) Corrosion in pipes having humid but gas free atmospheres; (2) Corrosion in pipes having humid atmospheres with measured quantities of oxygen, carbon dioxide and their mixtures; (3) Corrosion in pipes partly filled with water to determine whether greater corrosion takes place above or below the water level; (4) Corrosion in pipes entirely filled with dissolved gases.

G. L. Larson: I would like to call attention to the fact that this is the newest of our cooperative agreements with a well-known state institution. It is the type of agreement that is very beneficial to this Society because the Michigan School of Mines and Technology was very anxious to do research work along lines that would be of rapid benefit to the profession. We were able to arrange an agreement with them to the effect that we would furnish the technical advice and make available to them our source of publication and, in view of the fact that we are not able to furnish them any funds, they were very agreeable to that proposition.

Mr. Adler: That brings up the next point as to whether the free oxygen in the vapor above has any corrosive influence. In other words, if moisture is in the vaporous state and oxygen is dispersed in the mass, does corrosion take place? That is another point that might be investigated.

Mr. Walker: We have made numerous tests with specimens exposed partly to the steam and their accompanying gases and partly to condensation. We never have found any corrosion taking place above the water line. The corrosion in some cases seems to be more severe near the water line, probably due to the fact that the surface of the condensate is in contact with the gas and more gas is in solution there than farther down in the body of the condensate.

Another confirmation of the point is that in all of the cases of corrosion that we have seen, I do not recall any case where the corrosion was above the water line

in a pipe that was partly filled with condensate. It invariably is found in a channel or a series of pits below the water line.

T. M. DUGAN: I believe the Society is to be congratulated on their effort to secure data of this kind, inasmuch as the industry is suffering severely from the corrosive elements that enter into heating systems. No doubt, when we get real conclusions from Professor Seeber's paper, we will be better able to do proper engineering on systems from a corrosion standpoint.

No. 984

CARBON MONOXIDE SURVEYS OF TWO GARAGES

By A. H. Sluss,* (*NON-MEMBER*), Lawrence, Kan., E. K. Campbell † (*MEMBER*) Kansas City, Mo., and L. M. Farber ‡ (*NON-MEMBER*), Lawrence, Kan.

During 1932 and 1933 co-operative studies on the minimum heating requirements of garages, with provision for the same dilution of the carbon monoxide, have been conducted at Lawrence, Kan., and Kansas City, Mo., by the Kansas City Chapter of the American Society of Heating and Ventilating Engineers and the University of Kansas, in co-operation with the Research Laboratory of the A. S. H. V. E., and under the auspices of the Technical Advisory Committee on Ventilation of Garages and Bus Terminals.

Tests at Lawrence

The tests at Lawrence, Kan., were conducted in the Davis-Childs Chevrolet Garage, a single story building facing west with its south and east walls exposed. The building was 18 ft high, and was divided, as shown in Fig. 1, into a shop and a salesroom and office. Fig. 1 also shows the locations of the sampling stations.

Test Conditions and Results

Test L-1 was made February 29, 1932, under regular operating conditions, with doors, windows, and roof ventilators open, and with engine testing and general repair work being carried on. No carbon monoxide was found during this test.

Test L-2 was conducted under ordinary operating conditions on March 5, 1932, with doors and windows closed and with no special provision for ventilation. Engine testing and general repair work were being done at the time of the ventilation test, the results of which are shown in Table 1. It was found that carbon monoxide, too concentrated to be safe, was fairly evenly distributed throughout the shop and salesroom.

Tests L-3, L-4, L-5, and L-6 were conducted March 13, 1932, under special conditions to study natural ventilation. The results of these tests, shown in Table 2, indicate: (1) The results of tests L-2 and L-3 show that in garages with no mechanical ventilation, carbon monoxide tends to be uniformly dis-

* Professor of Mechanical Engineering, University of Kansas.
† Chairman, Technical Advisory Committee on Ventilation of Garages and Bus Terminals.
‡ Graduate Student, University of Kansas.
Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934, by A. H. Sluss.

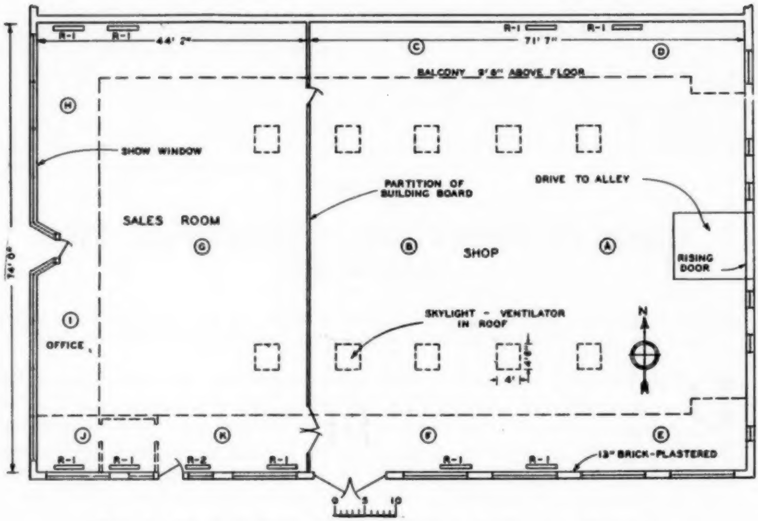


Fig. 1. Plan of Davis-Childs Garage, Lawrence, Kansas

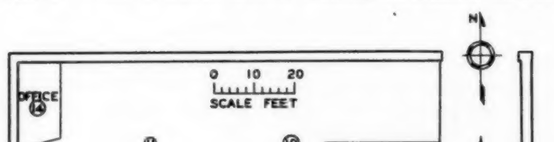


Fig. 2. Plan of United Motors Garage, Kansas City, Mo.

TABLE 1. TEST L-2. GENERAL SURVEY OF CARBON MONOXIDE CONCENTRATION IN LAWRENCE GARAGE

Date, March 5, 1932; time, 3 P.M. to 4 P.M.; outside temperature, 26 F.; inside temperature at 5 ft. level, 65 F; wind, 5 mph, N. W.; all garage ventilators and windows closed; activity in garage; general repairing; car movement in and out estimated at 4 cars per hour.

	CO Concentration—Parts per 10,000										
Sampling Stations and Locations / Elevation above Floor, Ft.	A East Center Shop	B West Center Shop	C N. W. Corner Shop	D N. E. Corner Shop	E S. E. Corner Shop	F S. W. Corner Shop	G Center Sales Room	H N. W. Corner Office	I West Center Office	J S. W. Corner Office	K S. E. Corner Office
1	0.8	1.8	3.2	1.8	1.3	0.8	1.8	0.0	...	1.3	1.3
5	0.8	1.8	1.8	1.8	1.3	1.8	1.8	0.8	1.8	1.8	0.8
10	0.8	1.3	1.8	1.8	1.3	0.8	2.6	1.8	1.3	1.8	0.8
15	...	...	...	...	...	...	...	1.8	1.8	1.8	...
18	1.3	1.3	1.8	1.3	1.8	1.8	1.8	...	...	...	...
25	1.3	1.3	...	...	...	...	1.8	...	...	...	...

tributed except where it is generated and in unswept portions of the garage; (2) When using roof ventilators, with no provision for admitting air, the carbon monoxide concentration was not reduced; (3) Under the same conditions, but with provision for allowing unheated air to enter, the carbon monoxide content was rapidly lowered, but the temperatures were reduced below the comfort range; (4) With three cars idling and with the roof ventilators and the side and rear doors open, the carbon monoxide concentration was reduced to a safe value.

TESTS AT KANSAS CITY

The tests at Kansas City were conducted in the garage of the United Motors Service Co., a two story building facing west with the north and east sides exposed. Only the first story, 14 ft high, was under observation. As shown in Fig. 2, an electric shop was located in a recess in its southeast corner, surrounded by solid walls on three sides, but with no intervening wall between it and the garage. The floor of this section was three feet higher than the main floor. Engine testing was ordinarily conducted on the main floor just north of the electric shop.

Since there was little air movement in the shop and its breathing line was three feet higher than that in the main room, carbon monoxide tended to concentrate in this space. In order to blow it out, a centrifugal fan type unit heater was installed as shown on the drawing, at 5, with a duct carrying outside air into the electric shop, the main discharge being horizontal toward the back wall. A study of the air movement resulting from this single opening showed that when the air was heated it did not stay at the floor long enough to accomplish its purpose, but rose and passed out near the ceiling. A small extension of the duct was therefore made to blow directly toward the floor. The total

TABLE 2. REPORT OF TESTS L-3, L-4, L-5, AND L-6. SURVEYS OF CARBON MONOXIDE CONCENTRATION IN LAWRENCE GARAGE WHEN USING ROOF VENTILATORS

Date, March 13, 1932; Outside Temperature, 35 F; Wind, N.N.W., 15 mph.

Test No.	L-3					L-4					L-5					L-6	
Time of Sampling	6:40–7:10 P.M.					8:05–8:20 P.M.					8:30–8:45 P.M.					8:53–9:00 P.M.	
Ventilation	No Ventilation					4 South and 1 Northeast Roof Ventilators Open from 7:15 P.M. to 8:20 P.M.					4 Roof Ventilators Open from 7:20 P.M. to 8:45 P.M. South Door Open from 8:22 P.M. to 8:45 P.M.					4 Roof Ventilators Open from 7:20 P.M. to 9:00 P.M. South Door Open from 8:22 P.M. to 9:00 P.M. East Door Open from 8:53 P.M. to 9:00 P.M.	
No. Cars Running	5					3					3					3	
	CO Conc. Parts per 10,000				Temp. F	CO Conc. Parts per 10,000				Temp. F	CO Conc. Parts per 10,000				Temp. F	CO Conc. Parts per 10,000	
Sampling Stations and Locations / Elevation above Floor, Feet	B W. Center Shop	D N. E. Corner Shop	E S. E. Corner Shop	F S. W. Corner Shop		B W. Center Shop	D N. E. Corner Shop	E S. E. Corner Shop	F S. W. Corner Shop		B W. Center Shop	D N. E. Corner Shop	E S. E. Corner Shop	F S. W. Corner Shop		B W. Center Shop	D N. E. Corner Shop
1	0.8	1.3	1.3	1.8	61	1.8	0.8	0.8	0.8	53	0.8	0.8	0.8	0.0	37	..	..
5	0.8	0.8	0.8	1.3	62	2.6	1.8	1.8	2.6	53	0.8	1.3	0.8	0.0	40	..	0.8
9	..	..	..	..	..	..	..	..	..	..	..	0.8	0.8	0.0	53	..	..
10	1.3	0.8	0.8	1.8	..	2.6	2.6	1.8	2.6	55	1.3	..	..	..	..	0.0	..
15	..	1.3	1.3	1.3	..	..	1.8	1.8	1.1	..	..	..	..	..	..	..	..
18	1.3	..	..	..	..	1.1	..	..	..	..	1.3	..	..	..	..	0.8	..
25	1.3	..	..	..	64	0.8	..	..	..	..	..	..	..	..	..	0.0	..

volume of air moved by the unit, on anemometer test, was 3600 cfm, which was a change of air for the entire electric shop about once every four minutes. Provision was made so air could be either recirculated or taken from the outside. During all of the tests, outside air was used. Owing to the mild winter, it was very difficult to find a cold day when the doors were kept closed and when business was heavy.

Test Conditions and Results

Test KC-7 was conducted on February 15, 1933, with no special provision for ventilation. Business was light and did not involve engine testing except for a quarter hour at 2:15 p. m. The results of this test, shown in Table 3, indicate that on a day when the doors must be kept closed and when business is light, the carbon monoxide concentration is kept at safe levels by infiltration and by air which enters when the doors are opened for cars.

TABLE 3. TEST KC-7

Time P.M.	Station	Height above Floor Feet	Parts CO in 10,000 Air	Remarks
2:00	1	5	...	Test spoiled
2:10	2	5	0.0	
2:13	5	5	0.0	
2:14	4	5	1.0	
2:15	3	5	0.2	
2:22	8	5	6.5	Cars running in front of electric shop
2:24	8	5	6.5	
2:26	8	½	1.6	
2:40	7	½	1.0	
2:40	7	5	1.0	
2:45	1	5	0.0	
2:45	1	1	1.0	
2:45	2	1	1.0	
2:48	2	5	1.0	
2:50	5	1	1.0	
2:50	5	5	1.0	
2:55	3	5	1.0	
2:55	3	1	1.0	
3:00	4	5	0.0	Small fan running on each side of sample
3:00	4	1	1.0	
3:05	11	5	0.0	
3:05	11	1	1.0	
3:10	9	5	1.0	Main door opened
3:10	9	1	0.0	

Tests KC-8, KC-9, and KC-10, were conducted March 11, 1933, to study the effect of the ventilating system on the distribution of carbon monoxide. Their results, given in Table 4, indicate that, taking the initial concentration and the interval between sampling into account, the system satisfactorily reduced the concentration of carbon monoxide in the electric shop and in the adjacent space. It was noticed that the air cleared first in the electric shop and that there was a movement of visible exhaust fumes into the main room. When the north windows were opened at the top to the same area as the air outlet of the ventilating system, the movement of the fumes was accelerated. The results of these tests indicate that the carbon monoxide concentration in a given space may be satisfactorily reduced by a ventilating system, but that definite provision must be made to remove the mixture which is driven away.

Reports of headaches began to come from the north side of the main room, where previously there had been no complaints, indicating a relatively higher concentration of carbon monoxide there because it was moved from the electric shop. It was found that the windows in the north side could be opened with no discomfort to the workmen, so that a draft diluted the carbon monoxide concentration to the point where reports of headaches ceased.

Test KC-11 was conducted on March 20, 1933, to determine the carbon monoxide concentration when using the ventilating system with the windows closed (except as noted in the foregoing paragraph) and the doors closed except for cars to enter or leave. Business during this test was considered heavy by the

TABLE 4. TESTS KC-8, KC-9, KC-10. SURVEYS OF CARBON MONOXIDE CONCENTRATION IN KANSAS CITY GARAGE USING MECHANICAL VENTILATION

Test No.	KC-8		KC-9		KC-10		
Time of Sampling	7:36–8:00 P.M.		8:15–9:14 P.M.		10:35–11:00 P.M.		
Ventilation	Fan Off Windows and Doors Closed		Fan Operating Windows and Doors Closed		Fan Operating Windows and Doors Closed		
Car Operation	Running		Running		Engines Stopped		
	CO Conc. Parts/10,000		CO Conc. Parts/10,000		CO Conc. Parts/10,000		
Elevation above Floor, Ft. / Sampling Stations and Locations	1	5	1	5	1	5	12
2. S. E. Shop	...	4.3	...	2.5	...	0.0	...
1. S. Center Shop	2.3	4.3	4.3	5.3	0.0	0.4	0.0
6. S. W. Shop	4.3	4.3	...	3.4	...	0.0	...
5. N. E. Shop	...	11.0	...	4.3	...	0.0	...
3. N. W. Shop	...	8.8	...	3.3	...	0.0	...
7. S. E. Service	...	10.0	...	4.2	...	0.0	...
8. S. Center Service	...	4.3	...	4.2	...	0.0	...
9. S. W. Service	...	4.3	...	5.3	...	0.0	...
12. E. Center Service	3.3	6.5	5.3	5.3	...	0.0	...
10. N. E. Service	...	1.8	...	...	...	0.0	...
11. N. W. Service	...	6.5	...	...	...	0.0	...

management of the garage. The doors were opened an average of twenty times an hour; the average number of cars in the garage was fifteen; and the rate of change was about fifteen cars per hour. It was estimated that during the test one car was idling all the time and one car was moving half the time. The results of the analysis in Test KC-11 are shown in Table 5, and indicate that the carbon monoxide concentration in the entire space had been reduced to a satisfactory amount. The clerks in the small office reported headaches, and it was found that at the time of this test the carbon monoxide in the office was one part in 10,000. As the office was supplied with excess radiation, a small fan was installed to furnish it with the necessary amount of fresh air.

CONCLUSIONS

The results of these studies show, in general, that the requirement of human comfort causes the need for ventilation, therefore:

(1) In most garages there is no carbon monoxide hazard, except in cold weather when it is necessary to close the doors and windows.

(2) Ventilation with unheated air is not feasible during cold weather.

Table 5. Test KC-11

Time	Station	Height above Floor, Ft.	Parts CO in 10,000 Air
8:55 A.M.	3	9	0.0
8:58	3	5	0.0
9:00	2	5	0.0
9:08	8	5	0.0
9:11	9	5	0.0
9:12	11	5	0.4
9:40	3	9	0.4
9:43	3	5	0.0
9:45	2	5	0.0
9:49	8	5	...
9:51	9	5	0.4
9:55	11	5	0.4
10:45	3	5	1.0
10:47	3	9	0.0
10:53	2	5	0.4
10:55	8	5	1.0
11:00	9	5	1.0
11:05	11	5	0.4
11:10	13	5	0.4
11:45	3	5	0.0
11:46	3	9	0.5
11:50	2	5	0.4
11:55	8	5	0.2
11:58	9	5	0.0
12:00	11	5	0.4
1:58 P.M.	11	5	0.2
1:59	9	5	0.2
2:00	8	5	0.2
2:02	3	5 & 3	0.4
2:03	2	5 & 3	0.0
Opened north windows 6 in. from top			
2:51	11	5	0.0
2:52	9	5	1.0
2:53	8	5	1.0
2:55	3	5 & 3	1.0
2:57	2	5 & 3	0.0
3:22	11	5	0.4
3:23	9	5	0.0
3:24	8	5	0.0
3:26	14	5	1.0
3:27	14	5	0.4
3:29	3	8	0.4
3:32	2	8	0.4
3:50	Outside	..	0.0

(3) Carbon monoxide concentration tended to increase in unswept portions of the room.

(4) The amount of air necessary to eliminate the carbon monoxide hazard in a garage may be reduced from the theoretical volume, providing it is properly introduced into the unswept portions, thereby driving the mixture into the swept portion where it can be disposed of by infiltration.

(5) The permissible reduction from the theoretical volume of 5000 cfm per idling car is in proportion to the volume of infiltration effectively utilized.

(6) Outside air used to dilute CO and sweep otherwise dead corners and pockets, must be heated sufficiently to maintain comfort, and it was found that extra heat introduced with the outside air allowed a considerable increase in infiltration without increasing discomfort.

(7) An interesting parallel is developing between CO and gasoline fumes, in that, under many conditions, the formation of pockets supplies the hazard and for the solution of either problem, the ventilation system must be so designed as to break up pockets and utilize the natural infiltration as well as the directly supplied and heated outside air, for dilution in order to keep the costs within commercial limits.

ACKNOWLEDGMENT

Acknowledgment is made of the cooperation of the following who have rendered valuable assistance in carrying on these studies: Davis Childs Chevrolet Co., Lawrence, Kan., United Motor Service Co., Kansas City Branch, Mr. Ward Cole, graduate student of Biochemistry of the University of Kansas, and Mr. John G. Lewis, Kansas City (Member).

DISCUSSION

E. K. CAMPBELL: When, at first, tests were conducted in the United Motors garage on the men working in the electrical shop, which was located in a small corner with the air cut off on three sides, the results indicated that carbon monoxide was present there. The workers in the electrical shop were the only ones who were particularly affected, and these men were being treated with aspirin, and it was necessary to take some of them home in the middle of the afternoon. Thus, in working out the solution to the problem, we were guided by the necessity of holding down the cost to the point where possibly the company could be persuaded to buy the apparatus if it operated successfully, and by the fact that the workers must be taken care of.

The apparatus was designed and installed to clear the electrical shop of carbon monoxide and, while Professor Sluss gives an air change of approximately four changes per hour for the entire area affected by the testing of cars, it was approximately double that for this particular section. The immediate result was that the carbon monoxide was driven across the room to the north side of the garage, between the doors, where it was eliminated by opening the doors and a central window on that side, without any additional apparatus.

The significant fact is that the unit heater provided sufficient heat, so that the occupants were willing to keep the doors and windows open a little longer than was absolutely necessary. I believe that any garage system which does not provide more than enough heat to maintain the proper temperature, will not be operated for ventilation.

Ventilation of a garage, as in any other type of building, is closely related to human comfort and whenever an uncomfortable condition is created by the ventilation system, I believe that the system will not be operated. The comfort of the workers should be given first consideration, if a garage ventilating system is to be operated successfully.

No. 985

DIURNAL AND SEASONAL VARIATIONS IN THE SMALL-ION CONTENT OF OUT-DOOR AND INDOOR AIR

By C. P. YAGLOU * (*MEMBER*) AND L. CLARIBEL BENJAMIN † (*NON-MEMBER*)
BOSTON, MASS.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers, and conducted at Harvard University, School of Public Health, Boston, Mass.

IN a former paper [1] it was shown that the small-ion content of air of occupied rooms is rapidly depleted after the occupants enter, and that normal ionization is not restored until after the occupants leave the room. The present report presents the results of observations on the diurnal and seasonal variations in the small-ion content of outdoor air, and on the corresponding changes which take place in lightly occupied rooms, heated and ventilated by the usual methods.

Measurements of atmospheric ionization have been made by many workers in various parts of the world, notably by the Carnegie Institute of Washington [2] during the cruises of the *Carnegie* in the Pacific, Atlantic, and Indian Oceans, and by Nolan and Nolan [3] at Glencree Valley, Ireland. In all instances the chief interest centered on the ionization condition of outdoor air with little or no regard to changes in the ionization indoors and to the possible relation of such changes to air conditioning.

SITUATION OF BUILDING AND ROOMS

The observations to be described were carried out at the Harvard School of Public Health which is about two miles west of the Atlantic Coastline and about the same distance southwest of the Weather Bureau Station, from which

* Assistant Professor of Industrial Hygiene, Harvard School of Public Health.
† Technician, Department of Industrial Hygiene.

[1] Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate, A.S.H.V.E. TRANSACTIOS, Vol. 38, 1932, p. 191.

[2] Atmospheric Electric Results Obtained Aboard the Carnegie, 1915-1921, by J. P. Ault and S. L. Mauchly, *Publication No. 175*, 1926, *5*, 195, Carnegie Institute, Washington, D. C.

[3] Observations on Atmospheric Ionization at Glencree, County Wicklow, by J. J. Nolan and P. J. Nolan, *Proc. Royal Irish Academy*, 1931, *40*, section A, No. 2.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by C. P. Yaglou.

some of the meteorological data not locally recorded were procured. The School is a three-story marble building with a flat cinder-fill roof. It has a basement floor partly below grade and a tunnel underneath the basement. It is surrounded by other school buildings and hospitals, all of which are supplied with heat and power from a central plant about 500 ft northwest of the recording station. Owing partly to this feature and partly to the unusually tall stack of the power plant, the air pollution in the vicinity of the recording station was not sufficient to affect the observations. However, during a brief

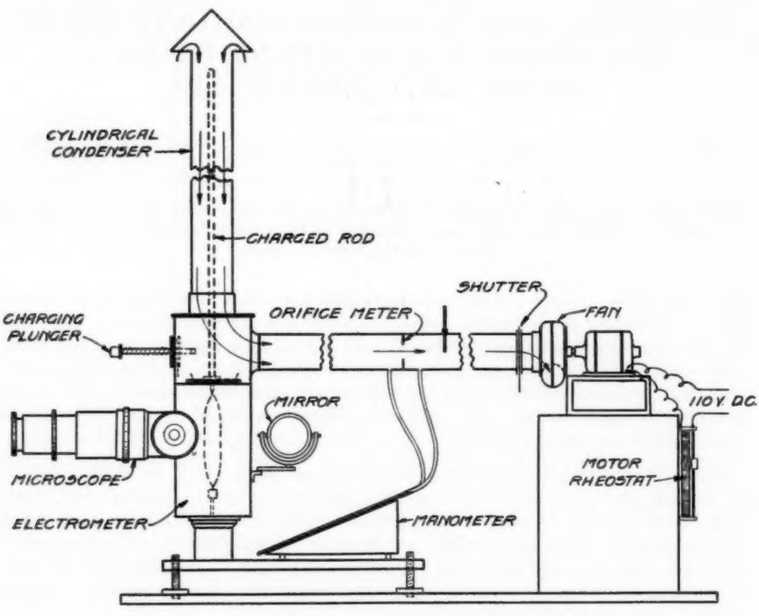


FIG. 1. APPARATUS FOR COUNTING IONS IN AIR

period of street repairs, a number of readings had to be rejected because of contamination by tar smoke.

During the first year and a half (May, 1930, to Sept., 1931) the observations were made in a reading room, 24 x 15 x 8 ft, on the third floor of the building, having three large windows on a southwest wall and four doors. Two of the doors open onto a terrace, a third opens into a connecting room, and the fourth into a main corridor. The room is heated by two steam radiators, located under the windows and ventilated by opening windows and doors. Besides the observer there were usually one or two persons in the room when the measurements were made, and the occupants were at liberty to open or close the windows and doors and come and go as they pleased. An undesirable feature of this location was that the air in the room was at times vitiated by

gases from laboratories across the hall. Some of the indoor readings were thus affected and had to be discarded.

From September, 1931 to May, 1933, the observations were continued in a smaller room, 10 x 12 x 12 ft on the second floor under more or less controlled conditions. This room has two windows on a southeast wall and one door. It is used solely by the observer as an office and observation room. There are no laboratories on this floor, but in order to guard against the possibility of vitiation from a classroom diagonally across the hall, the door of the observation room was kept shut. The air conditions in the room were controlled by

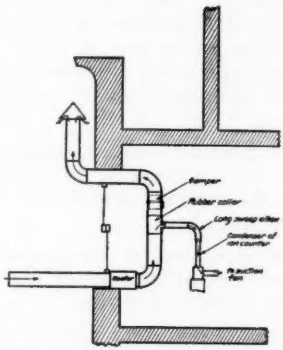


FIG. 2. METHOD OF PASSING OUTDOOR AIR THROUGH ION COUNTER

opening or shutting the windows and radiator valve so as to suit the comfort of the observer.

SPECIFICATIONS OF ION COUNTERS

The apparatus used for counting the number of small ions per cubic centimeter of air has been described in a previous paper,[1] and a drawing of it is shown in Fig. 1. Following are specifications of the instruments which were not given in the earlier report.

The electrometers were specially made, double-fiber, portable, Wulf type, capable of measuring a maximum potential of 100 volts. In order to minimize electrical leakage and polarization, a metal guard ring was cast in the amber insulation disc, and the ring was charged to the same potential as the collecting electrode by means of a changing plunger (see Fig. 1). The sensitivity of the instruments was between 1 and 2 divisions per volt, and they could be easily read to one-tenth of a division or less, with a magnification power of about 100. The normal charge applied to the system was 92 volts (2 radio *B* batteries) and the normal airflow (measured by means of a calibrated orifice meter) was 143 liters per minute. Since the effective length of the collecting electrode was 37 cm and the condenser outer tube 3 cm in diameter, theoretically

the ions caught had mobilities greater than 0.2 cm per second and per volt per centimeter.

All data presented herein refer to this particular group of small ions.

METHOD OF MAKING OBSERVATIONS

Fig. 2 shows the arrangement used for drawing samples of outdoor air through the counters without having to take the instruments out into the open. Outdoor air was circulated through a 6-in. galvanized iron duct, the two ends of which projected to the exterior through boards fitted in the window openings. The inlet of the duct (lower end) was 38 ft above ground in the observations made on the third floor, and 22 ft in those taken on the second floor. In cold weather the air circulated by gravity and the temperature of the sample was kept close to 70 F (room temperature) by means of an electric heater at the base of the sampling duct. In warm weather circulation was effected by means of a suction fan.

Simultaneous ion counts in the air stream inside the duct, and on the terrace, under a tent, at a point about 6 in. from the inlet to the duct, showed no appreciable difference in the results. Similar tests with the electric heater on and off showed no apparent effect.

It is believed that the upper end of the sampling duct was sufficiently protected by the walls and eaves of the building against possible direct action of the earth's field, known as *electrode action.*[4]

Samples of outdoor air were passed through the counter by connecting the upper end of the condenser tube to the sampling duct with a long-sweep elbow (see Fig. 2). Originally the elbow was made of glass tube and it proved perfectly satisfactory, but after a year it was found that in very cold and dry weather, the glass adsorbed some of the ions passing through it. It was therefore, replaced with a brass elbow which gave satisfactory results regardless of humidity or temperature. An attempt was made to establish by special tests correction factors for the adsorption at various humidities, but no exact relationship could be found. Fortunately this error did not affect the indoor readings because the elbow had to be removed for drawing room air through the counter.

Systematic counts of small ions were made daily (Sundays and principal holidays excluded) from May, 1930 to May, 1933, at the hours 9:00-11:00 a. m. (75th meridian time), 1:30-2:00 p. m., and 4:30-5:00 p. m. for the ionization outdoors and at 9:00-10:00 a. m. and 2:00-2:30 p. m. for the ionization indoors. These hours were chosen from studies of diurnal variations so as to make the average of the three day-time readings (two for indoor air) approximately equal to the 24-hour average of the diurnal variation tests, in which observations were made every hour.

Prior to September, 1931, a single counter was used for both positive and negative ions, by reversing the charge on the collecting electrode. This proved unsatisfactory for many reasons and since September, 1931 two identical counters were used simultaneously, one for positive and one for negative ions,

[4] The Electrical Conductivity of the Atmosphere and Its Causes, by V. F. Hess, New York, D. Van Nostrand & Co., 1928.

keeping them always charged with the same polarity. Before beginning observations the instruments were allowed to run for about 5 min until a steady state was reached. Each observation took five minutes. As a rule, two or three 5-min readings gave a good average, but in changeable weather more readings were necessary.

Owing to the low voltage charge (92 volts), the guard ring in the insulation, and the artificial heating of air in damp weather, the insulation leakage was

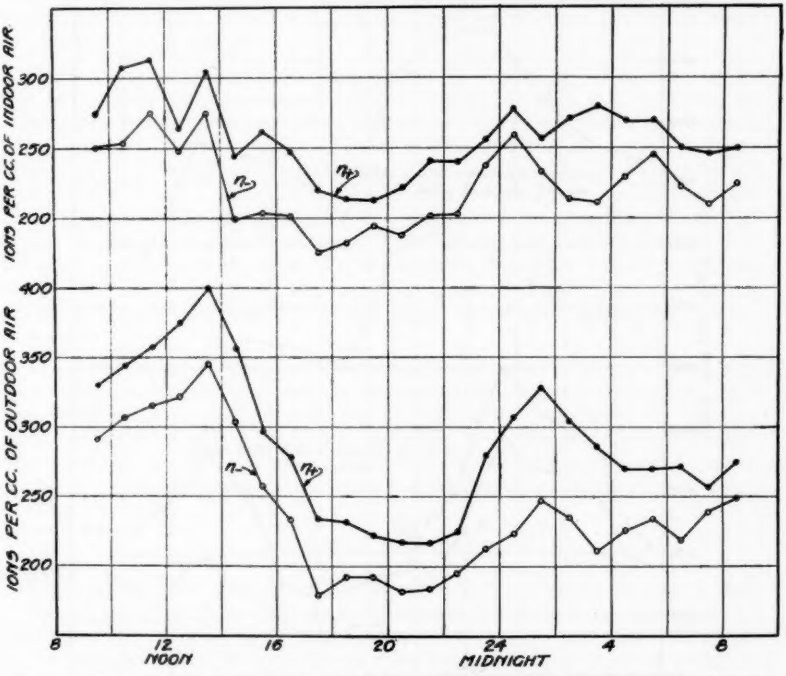


FIG. 3. DIURNAL VARIATION IN THE SMALL-ION CONTENT OF OUTDOOR AND INDOOR AIR IN CLEAR WEATHER, BOSTON, MASS., SEVENTY-FIFTH MERIDIAN TIME

nil except in unusually warm and rainy weather when it was determined and the correction, if any, applied to the readings.

In addition to a standard group of small ions (mobility upward of 0.2 cm per second per volt per centimeter) parallel counts were made of the so-called *molecular* ions,[4] having mobilities of 1 cm per second per volt per centimeter or higher, and from time to time counts were made of a broader group of ions, down to 0.03 cm per second per volt per centimeter mobility, by utilizing Ebert's standard ion counter. These observations were made for the purpose of estimating the extent to which *molecular* and intermediate ions were present in indoor and outdoor air. They will be presented in another paper.

DIURNAL VARIATION IN THE SMALL-ION CONTENT OF THE ATMOSPHERE

Studies of complete diurnal variation in the small-ion concentration of outdoor and indoor air were carried out once each season during 1930 and 1931, and once every month in 1932 and 1933. In this series of tests, observations were made every hour or oftener, beginning at 9:00 a. m. (75th meridian time) and continuing over a period of 24 hours or longer. In order to avoid

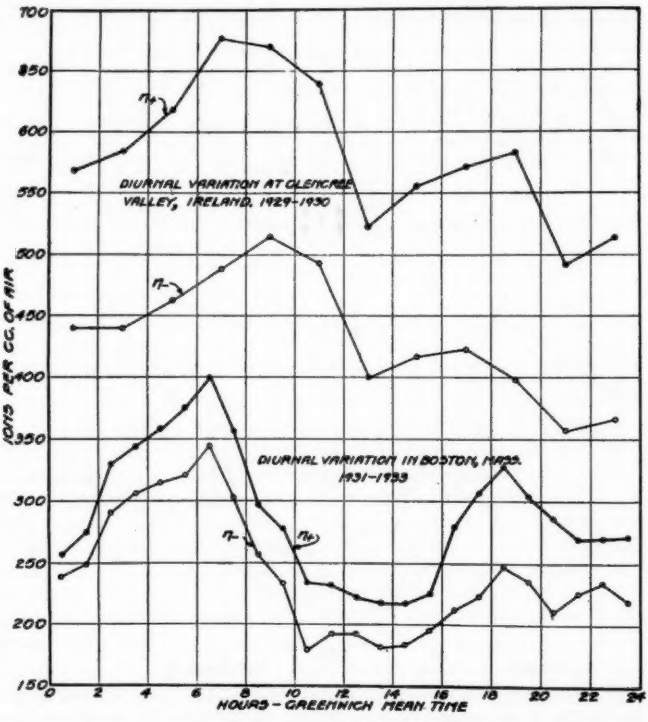


FIG. 4. COMPARISON OF DIURNAL VARIATION IN BOSTON, MASS., WITH THAT AT GLENCREE VALLEY, IRELAND, AS REPORTED BY NOLAN & NOLAN [3]

the disturbing influence of clouds, rain, snow, etc., tests were run in clear weather only. There were, however, several experiments in which unexpected cloudiness, rain, or snow near the middle or end of the run made the results unsuitable for the purpose. Omitting all such tests and all observations with the glass elbow, there remain thirteen good 24-hour experiments to be considered here, spread over the period of December, 1931, to May, 1933.

The hourly means of all thirteen diurnal variation runs are plotted in Fig. 3. The curves for outdoor air show two peaks and two valleys. The peaks occur

shortly after mid-day and mid-night; the valleys between 6 and 10 p. m. and 4 and 8 a. m. These characteristics persist more or less, in the individual runs and in the averages by season, the main difference being that the actual concentration of ions is higher somewhat in summer than in winter.

In indoor air the diurnal changes in ion content are less pronounced than in outdoor air (see Fig. 3), both the amplitude of variation and the actual ion content being lower. During the hours of maximum ionization the concentration of ions indoors (in a room occupied by one person) is about 20 per cent lower than that outdoors while during the hours of minimum ionization, there is practically no difference.

If the curves for outdoor air are plotted to Greenwich mean time as in Fig. 4, a fairly good agreement appears to exist between the ionization in Boston and that at Glencree Valley, Ireland, as reported by Nolan and Nolan.[3] The curves are somewhat similar in shape and the maxima and minima occur at approximately the same universal time. This supports the prevailing belief that the sun is not a sufficient nor an important direct cause of atmospheric ionization, since both at Glencree Valley and Boston the maxima and minima appear to occur at practically the same universal time in spite of the fact that there is a difference of about 5 hours between the local times of the two places. In other words, while the ionization in Boston is found to be somewhat higher during the day than during the night, at Glencree Valley the reverse holds true.

The difference between the concentration of ions at Glencree Valley and Boston is probably real, and may be ascribed to differences in topography and air pollution. By contrast with the thickly settled community of Boston, Glencree Valley is thinly populated, and direct contamination from chimneys and automobile exhausts is small. Observations made by us in country places within 25 miles of Boston gave results comparable or even higher than those of Glencree Valley.

SEASONAL VARIATIONS

The monthly daily average small-ion content of air shown in Fig. 5 was computed from the three daily sets of readings taken in the forenoon, early afternoon, and early evening. Each point on the curves represents the mean of between 22 and 28 daily averages; each daily average is in turn the mean of three sets of readings (two for indoor air), each set consisting of two or more ion counts. The data include all day-time observations made at the specified hours, except those vitiated by smoke (nine sets altogether), and those affected by heavy rains (15 sets) to such an extent as to show concentrations of 500 or more ions per cubic centimeter of air.

The curves for each of the three years agree well among themselves in showing distinct minima during the winter months and maxima during the summer months. The dotted section of the curves for the winter of 1930-31 represents doubtful data owing to a probable adsorption of ions by the glass elbow. The winter of 1930-1931 was the coldest and that of 1932-33 the warmest of the three, but all three winters were considerably milder than the average for Boston. Therefore the winter results may not be representative of normal weather for the locality.

Under the conditions which existed in the observation room, namely with the observer alone in the room, and no smoking, the mean seasonal curves for outdoor and indoor air (Fig. 5) do not differ radically except in the ampli-

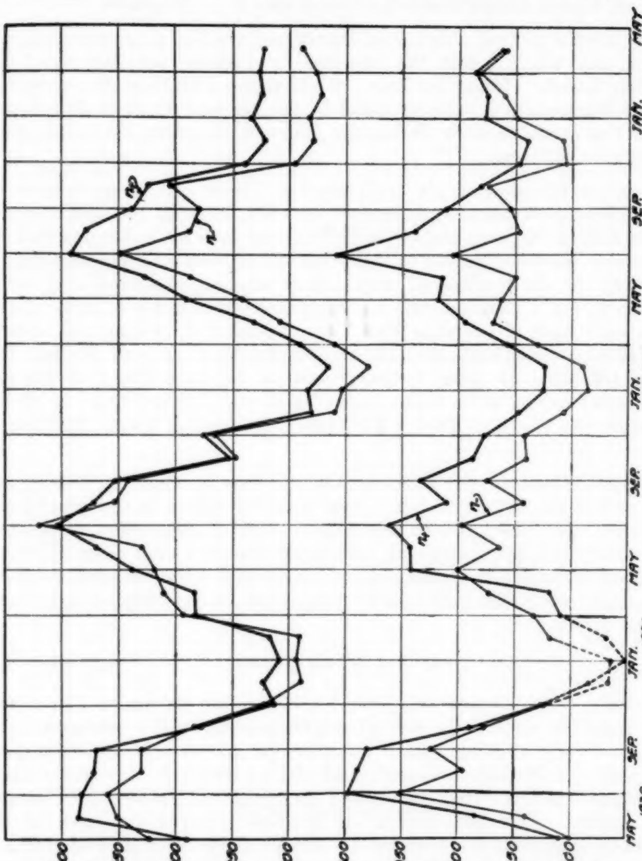


Fig. 5. Seasonal Variation in the Small-Ion Content of Outdoor and Indoor Air, Boston, Mass.

tude of variation from winter to summer. Omitting values for the winter of 1930-31, the ionization during the winter and spring of 1932 and 1933 appears to be lower indoors than out-of-doors; in the autumn season the difference becomes less noticeable whereas in the summer the ionization indoors is on the whole higher than that outdoors.

In winter when the windows of the observation room were mostly shut, marked changes in outdoor ionization were accompanied with comparatively small changes indoors (see Fig. 6).

In summer such contrasts did not usually occur, and the full range of outdoor changes in ionization was realized indoors (Fig. 6), except under certain conditions of humidity, cloudiness, and rain (dealt with further on) which produced a distinct lowering in the outdoor concentration of small ions without affecting appreciably the concentration indoors. Even in clear summer weather the small-ion content of indoor air was often higher than that of outdoor air. On the other hand, the *molecular* ions, which are most sensitive of all ions to changes in air conditions, were at all seasons less numerous in indoor than in outdoor air.

The only data available for comparison of the seasonal variation curves for outdoor air are certain observations [5] of the Carnegie Institute of Washington made over a period of four months, December, 1915, to March, 1916, during the cruises of the *Carnegie* in the Pacific, Atlantic, and Indian Oceans. Minimum ionization for these three months occurred in January as in our observations, but Wait [6] does not believe that the data over the sea are sufficient to warrant any conclusions.

The Glencree Valley observations are not, according to Nolan and Nolan [3] well enough distributed over the 12 months of the year to conclude whether there is any annual variation in the ionization.

VARIATION WITH TEMPERATURE AND HUMIDITY

It is generally believed that there is a direct relationship between temperature and atmospheric ionization, the connection being sometimes attributed to an ionizing influence of the sun. Table 1 presents the results of an examination for possible association with temperature and humidity at like seasons. In order to preclude any possible unbalance in the data, all readings taken by the use of the glass elbow (those prior to May, 1931) were rejected. Likewise the observations made on days with precipitation of 0.01 in. or more between 8 a. m. and 8 p. m., were excluded. The figures in the brackets shown in Table 1 represent the number of daily averages from which the mean ion numbers were derived.

For temperatures under 70 F a rise in temperature appears to be associated with an increase in the ion numbers at all seasons and all percentages of relative humidity (Table 1). However, this direct association does not hold for temperatures over 70 F. The numerous observations in the summer season indicate a decrease in ionization when the temperature exceeds 70 F.

More definite than the effect of temperature is the inverse relationship between humidity and ionization (Table 1). With a rising humidity, the concentration of ions decreases consistently at all temperatures and seasons. The decrease seems to be largely a pure humidity effect as in about two-thirds of the cases it was not associated with cloudiness on the day of observations nor with rain or cloudiness within 24 hours following the observations.

---

[5] Results of Atmospheric Electric Observations Made Aboard the Galilee (1907-1908), by L. A. Bauer and W. F. Swann, and *Publication No. 175, Vol. 3,* Carnegie (1909-1916). Carnegie Institute, Washington, D. C.

[6] Variations in the Small-ion Content of the Atmosphere and Their Causes, by G. R. Wait, *Journal Franklin Institute,* 1933, *216,* 147.

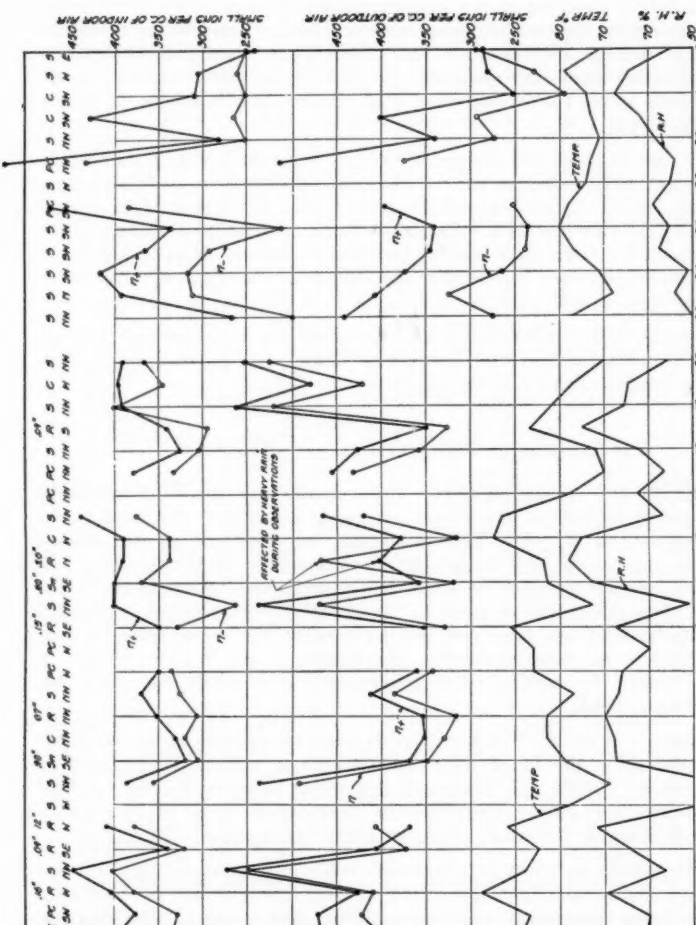


FIG. 6. INTERDIURNAL CHANGES IN TEMPERATURE AND HUMIDITY IN RELATION TO CHANGES IN THE SMALL-ION CONTENT OF OUTDOOR AND INDOOR AIR

Legend: Precipitation: 8 a. m. to 8 p. m. Weather: *s*, clear; *pc*, partly cloudy; *c*, cloudy; *r*, rain; *sn*, snow. Wind: N, north; E, east; S, south; W, west.

So far as temperature and humidity are concerned, low concentrations of small ions appear to occur in moist air at temperatures under 50 F. (Table 1). The minimum concentrations fall in winter. High ion concentrations may

TABLE 1. DAILY AVERAGE SMALL-ION CONTENT OF OUTDOOR AIR IN RELATION TO TEMPERATURE AND HUMIDITY MAY, 1931 TO MAY, 1933. BOSTON, MASS.

(Excluding days with precipitation of 0.01 in. or more. Figures in brackets represent number of days.)

Mean Daytime Temperature F	Average Relative Humidity, Per Cent							
	35 or Less		36–55		56–75		76–100	
	n+	n−	n+	n−	n+	n−	n+	n−
Winter								
Under 32	323	289 (9)	288	254 (15)	265	232 (9)	179	150 (2)
32–50			335	305 (13)	268	231 (36)	177	146 (7)
51–70					304	256 (4)	240	180 (1)
71 and over								
Summer								
Under 32								
32–50								
51–70			440	356 (12)	353	276 (27)	256	206 (7)
71 and over			403	285 (32)	334	238 (30)	262	231 (6)
Spring								
Under 32								
32–50	496	489 (3)	321	294 (15)	280	256 (15)	216	181 (7)
51–70	297	243 (5)	386	349 (24)	318	276 (16)	259	228 (5)
71 and over	470	373 (2)	351	263 (9)	414	305 (3)		
Autumn								
Under 32			278	238 (5)	251	222 (5)		
32–50	277	240 (2)	316	280 (17)	263	214 (26)	206	168 (6)
51–70			347	289 (11)	322	281 (30)	263	232 (11)
71 and over					307	220 (2)	356	299 (2)

occur at any temperature so long as the humidity is well under 75 per cent. The most frequent occurrence is in the summer when the temperature is between 60 and 70 F and the relative humidity between 35 and 55 per cent.

Much more important than either temperature or humidity appear to be the interdiurnal *changes* in temperature and humidity which in the absence of serious disturbances by rains or other causes are usually accompanied with inverse changes in ionization. Illustrations for winter and summer appear in

TABLE 2. EFFECT OF CLOUDINESS AND PRECIPITATION ON THE SMALL-ION CONTENT OF OUTDOOR AIR MAY, 1931 TO MAY, 1933. BOSTON, MASS.

([a] Excluding days with precipitation of 0.01 in. or more)

Character of Days	Winter n+	Winter n−	Winter n+/n−	Winter No. of Days	Spring n+	Spring n−	Spring n+/n−	Spring No. of Days
Clear days[a]	310	276	1.18	55	373	308	1.16	64
Partly cloudy[a]	260	226	1.22	27	282	248	1.18	30
Cloudy[a]	188	143	1.25	14	207	181	1.19	10
Character of Precipitation	**n+**	**n−**	**n+/n−**	**Sets of Observations**	**n+**	**n−**	**n+/n−**	**Sets of Observations**
Fog	195	155	1.25	3	138	144	0.98	3
Light rain	190	176	1 10	18	173	154	1.19	22
Moderate rain	195	224	0.92	24	193	208	1.00	7
[b] Heavy rain	205	254	0.89	4	342	574	0.70	7
Heavy rain with thunder					1342	1465	0.92	1
Snow					181	173	1.04	2
Heavy snow	291	325	0.89	1				

Character of Days	Summer n+	Summer n−	Summer n+/n−	Summer No. of Days	Autumn n+	Autumn n−	Autumn n+/n−	Autumn No. of Days
Clear days[a]	379	286	1.39	67	316	268	1.22	55
Partly cloudy[a]	350	272	1.31	38	302	260	1.21	32
Cloudy[a]	288	230	1.35	9	241	201	1.22	30
Character of Precipitation	**n+**	**n−**	**n+/n−**	**Sets of Observations**	**n+**	**n−**	**n+/n−**	**Sets of Observations**
Fog	162	68	2.38	1	213	146	1.46	2
Light rain	297	226	1.33	7	187	166	1.20	18
Moderate rain	150	143	1.24	2	202	223	0.97	16
[b] Heavy rain	366	547	0.66	7	487	1032	0.47	7
Heavy rain with thunder					1342	1465	0.92	1
Snow					181	173	1.04	2
Heavy snow	291	325	0.89	1				

[b] No thunder.

Fig. 6. The temperature and humidity values shown are daytime averages corresponding to the time at which the ion counts were made. It makes no difference if the 24 hour averages are used instead. The breaks in the ionization curves occur on Sundays when no observations were made. It can be seen that a significant rise in the interdiurnal temperature or humidity is as

a rule preceded or accompanied with a falling ion content but if the temperature continues to rise for three days or more, the ionization begins to increase.

Reversely when the temperature or humidity falls, the small-ion content of air usually increases on the day of fall or on the preceding day, but if the fall continues for three days or more, the ionization begins to decrease. With few exceptions these changes persist regardless of actual temperature level, and it seems to make little difference whether the rise or drop of temperature brings pleasant or unpleasant weather.

In winter a decrease in the small-ion content can be more directly associated with approaching rain or snow storms than with a rise in temperature and humidity which usually precede such storms. Conversely, an increase in the ionization may be ascribed to the advent of clear weather, rather than to a drop in temperature and humidity at the end of a storm. In summer however, when storms occur much less frequently, the factor of precipitation does not seem to offer a satisfactory explanation for the inverse relationship between changes in temperature and humidity and changes in ionization. The data shown in Fig. 6 will bear out these statements.

The foregoing alterations in the small-ion content of the atmosphere are not entirely confined to outdoor air as they also affect the indoor air (Fig. 6) much more in the summer than in the winter as explained earlier.

INFLUENCE OF CLOUDINESS AND PRECIPITATION ON THE SMALL-ION CONTENT OF THE ATMOSPHERE

Small patchy clouds, other than those associated with precipitation, do not in general produce conspicuous effects on the small-ion content of air. On the other hand, the great majority of clouds which cover the sky wholly or partly, diminish considerably the ion concentration, as shown in Table 2. The effect is largely confined to outdoor air, except in overcast weather which eventually lowers the indoor concentration to that outdoors.

Light or moderate precipitation of all kinds produces a significant reduction in the small-ion numbers at all seasons (Table 2). Heavy precipitation results in a considerable increase of the number of negative ions which may be explained by the well-known Lenard[4] effect (p. 63). The increase appears to be much greater during showers and electrical storms than during steady rains, and it is least conspicuous in heavy rains of winter.

During a local thunderstorm in which heavy rain was accompanied with almost continuous thunder and lightning for about 20 min, the number of both positive and negative ions increased enormously, reaching peak values of 3000 and 3150 ions per cubic centimeter respectively at the height of the storm. The average values during the 20-min period were 1342 and 1465 for the positive and negative ions, as shown in Table 2. Despite this enormous increase in the outdoor concentration, the indoor concentration was but slightly increased.

The influence of precipitation on the small-ion content of indoor air depended a great deal upon the duration and character of the precipitation and to

a small extent upon the direction of the wind. Light and moderate precipitation of a few hours' duration did not as a rule affect appreciably the concentration of ions indoors. As a result the indoor air was richer in small ions than that

TABLE 3. VARIATION OF DAILY AVERAGE SMALL-ION CONTENT OF OUTDOOR AIR WITH BAROMETRIC PRESSURE MAY, 1931 TO MAY, 1933. BOSTON, MASS.

(Excluding days with precipitation of 0.01 in. or more. Figures in brackets represent number of days.)

Mean Daytime Temperature F	Average Barometric Pressure, In. Hg							
	29.75 or Less		29.76–30.00		30.01–30.25		30.26 and Over	
	n+	n−	n+	n−	n+	n−	n+	n−
Winter								
Under 32	293	269 (8)	286	252 (8)	303	274 (9)	264	216 (10)
32–50	316	292 (8)	281	239 (18)	276	245 (18)	221	196 (12)
51–70	308	265 (2)	187	137 (1)	327	269 (2)		
71 and over								
All temperatures	304	279 (18)	279	243 (27)	288	251 (29)	241	206 (22)
Summer								
Under 32								
32–50								
51–70	362	287 (7)	402	332 (18)	321	248 (16)	332	240 (5)
71 and over	379	286 (8)	370	272 (46)	315	223 (13)	320	213 (1)
All temperatures	317	286 (15)	379	289 (64)	318	236 (29)	330	236 (6)
Spring								
Under 32								
32–50	323	294 (6)	353	331 (11)	270	249 (16)	269	227 (7)
51–70	393	382 (5)	350	310 (19)	342	297 (23)	216	165 (3)
71 and over	431	326 (1)	328	237 (6)	420	325 (7)		
All temperatures	361	333 (12)	347	304 (36)	325	284 (46)	253	214 (10)
Autumn								
Under 32	268	277 (1)	220	177 (2)	202	211 (1)	289	243 (6)
32–50	347	274 (2)	302	255 (11)	299	254 (17)	232	197 (21)
51–70	304	265 (4)	372	328 (14)	285	241 (26)	320	274 (8)
71 and over			301	210 (1)	342	276 (3)		
All temperatures	311	269 (7)	330	284 (28)	292	248 (47)	262	222 (35)

outdoors before and during precipitation of such character, especially in the summer.

Likewise heavy showers failed to produce indoors the conspicuous increase in the number of negative ions noted out-of-doors, although in some instances there was a definite increase.

Persistent rain or snow tended to equalize the concentration of small ions in outdoor and indoor air. Wind blowing against the window openings of the observation room had a similar tendency, but the response was usually slow and gradual. A much more prompt response resulted when a current of outdoor air was blown into the room by means of a fan on the window sill. As soon as the room dry- and wet-bulb temperatures approached the outdoor values, the concentration of ions indoors fell substantially to the prevailing outdoor values.

VARIATION WITH BAROMETRIC PRESSURE AND WIND

Referring to Table 3, a rise in the barometric pressure appears to be associated with a decrease in the outdoor ion count, but the range of variation

TABLE 4. VARIATION OF DAILY AVERAGE SMALL-ION CONTENT OF OUTDOOR AIR WITH PREVAILING WIND DIRECTION MAY, 1931 TO MAY, 1933. BOSTON, MASS.

(Excluding days with precipitation of 0.01 in. or more)

Prevailing Wind	Winter			Spring			Summer			Autumn		
	n+	n−	No. of Days	n+	n−	No. of Days	n+	n−	No. of Days	n+	n−	No. of Days
N.........	188	145	7	191	159	1	370	279	3	247	210	6
NE & E...	224	186	3	226	188	16	265	214	14	262	219	6
SE & S....	...	...	..	320	275	8	342	258	18	255	238	9
SW.......	229	195	16	292	247	19	298	214	24	268	219	28
W.........	291	245	25	365	322	26	390	301	17	308	266	31
NW.......	306	272	45	380	341	34	429	326	38	316	280	37

is not great. An effect of this kind is to be expected if the ionization in the atmosphere near the surface of the earth is derived mainly from emanation of radio-active substances in the soil (60 per cent of it, according to Millikan [7]). A lowering of the pressure would cause soil air rich in emanation to be drawn up through the capillaries of the earth into the atmosphere, whereas a rising pressure would impede diffusion of this highly ionized air into the atmosphere above.

It is of interest to note that when the results are arranged according to temperature and barometric pressure, as in Table 3, the apparent connection between actual temperature and ionization observed in Table 1 is practically eliminated.

The variation with wind direction is shown in Table 4. The highest ionization values were obtained with winds from the West and Northwest, and the lowest with those from the Northeast and East. For the locality of Boston, West and Northwest winds usually bring clear weather, sharp drops in temperature, and comparatively low relative humidities. All of these conditions

[7] On the Question of the Constancy of the Cosmic Radiation and the Relation of These Rays to Meteorology, by R. A. Millikan, *Physical Rev.*, 1930, *36*, 1595.

were shown to be associated with high concentrations of small ions. On the other hand, winds from the East and Northeast bring overcast weather, and they are largely responsible for the storms of winter. As a rule they are preceded by a rise of both temperature and humidity, culminating in rain or snow. These adverse conditions find a powerful reflection in the small-ion content of outdoor air as has been seen.

Winds from other directions (North, South, Southeast and Southwest), when not accompanied by precipitation, give intermediate values between the maxima and minima, depending largely upon the humidity, cloudiness, and upon the drop or rise in temperature and humidity which accompany the wind.

The influence of wind direction upon the indoor concentration of ions is likewise an indirect one, associated with temperature and humidity changes, cloudiness, and precipitation, etc. which may or may not affect appreciably the indoor ions, as pointed out earlier.

Concerning wind velocity, the results are not very definite. With increasing force of wind from the West and Northwest, the concentration of outdoor ions is somewhat increased, owing perhaps to the drawing of strongly ionized air from the capillaries of the soil by the aspirating action of wind. Winds from the North, South, Southeast, and Southwest fail to show such an effect, while with winds from the East and Northeast the concentration of ions appears to decrease with increasing wind velocity.

A de-ionizing action with increasing force of winds from the East and Northeast is perfectly feasible when the ground is wet, as water in itself is an excellent ion trap, and the higher the wind velocity, the greater the diffusion of atmospheric ions to the water.

SUMMARY AND CONCLUSIONS

Daily observations of the small-ion content of outdoor and indoor air from May, 1930 to May, 1933, disclose definite diurnal and seasonal variations depending largely upon local and general meteorological conditions.

The most important climatic factors affecting the small-ion content of outdoor air appear to be the interdiurnal changes of temperature and humidity. A drop in the interdiurnal temperature and humidity is as a rule preceded or accompanied with a sharp rise in the ion content of air, and vice versa, provided that the drop or rise in temperature and humidity does not continue more than 2 days.

Cloudiness, high humidities, and light or moderate precipitation of any kind have a detrimental effect on the small-ion content of outdoor air. Heavy precipitation results in a considerable increase in the number of small negative ions, and when the precipitation is accompanied with thunder and lightning both positive and negative ions attain very high values.

In the locality of Boston, high concentrations of small ions are as a rule associated with winds from the W. and N. W., which usually bring sharp drops in the interdiurnal temperature and humidity and clear weather. The actual temperature and barometric pressure are not very important so long as the relative humidity is well under 75 per cent.

Low concentrations of small ions are generally associated with stormy winds from the E. and N. E., which are often preceded with a rise of temperature and humidity finally culminating in rain or snow. The minima occur in damp or moist weather when the temperature is under 50 F.

In winter, the concentration of small ions in indoor air is considerably lower than that in outdoor air. In summer the reverse seems to hold true. Adverse weather of short duration does not conspicuously affect the indoor concentration in spite of powerful effects on the outdoor ions. Persistent bad weather tends to equalize the outdoor and indoor ion numbers.

In the light of the present study it would seem that the small ion content of the atmosphere is an important meteorological factor which deserves wider recognition and more study than heretofore afforded to it.

ACKNOWLEDGMENT

The authors are greatly indebted to Sarah P. Choate and Allan D. Brandt of the Department of Industrial Hygiene for their valuable assistance in carrying out special observations.

DISCUSSION

C.-E. A. WINSLOW: Professor Yaglou's paper is of particular interest to us because we have done work of this kind at the Pierce Laboratories. We have only a short series of observations so far, but we have done our work with this difference: we have determined not only the small ions but also the large ions, divided into 4 classes as to size and mobility. Our results, so far as the small ions are concerned, coincide almost exactly with those of Professor Yaglou.

On the other hand, it is interesting to notice that whereas one gets from 2 to 600 of the small ions per cubic centimeter, there are from 10 to 20,000 of the large ions. They are much more numerous, and at this season when the small ions are predominantly negative, there is a slight predominance of positive ions of the larger size.

The relation between the total positive and negative charges seems to be as one might expect, almost one of equivalence, but there is a wide variation in the total ionic content, a variation perhaps at this season from 10,000 ions up to 20,000 of all sizes. There is also variation from day to day in the division of these ions according to their size. On some days the proportion of the small ions relative to the large ones will be very different from the proportion on other days.

There is also the additional variable of the relation between the positive and negative signs in the small ions and the large ones respectively. We are evidently getting into a very complicated and interesting subject and in some respects it seems probable that the large ions may be of just as much interest, perhaps more, than the small ions. In the first place, they are 10 to 100 times as numerous and according to the assumption of the German investigators they are of greater physiological importance because as a result of their lesser mobility they are carried further down into the respiratory tract. As I say, we are at the threshold of information which may prove to have considerable significance.

DR. C. A. MILLS [8]: For several years I have been greatly interested in the stimulation brought by our storms, particularly the cold polar waves that sweep down

[8] Professor of Experimental Medicine, University of Cincinnati.

out of the Canadian Northwest, sometimes as far as Texas, then turn back up northeastward to pass out over the Gulf of St. Lawrence. These cold heavy air masses passing down and across the continent mean clear cold bracing weather. Such air exhilarates us, acting like a sparkling champagne, and provides, I believe, one of the major features in the climatic stimulation to which we are subjected. More will be said on this score in my own paper a little later, but I wish just now to point out the sharp step-up in the numbers of both positive and negative ions which has just been shown to take place as these cold waves come along. Perhaps it is these ions, fresh from the upper atmosphere, where the air molecules have been bombarded by cosmic rays or other radiations, that are responsible for our stimulation.

We also need to know more about the physiologic effects of these storm changes on man, so that we may apply intelligent control. I have found, for instance, that suicides come in waves with rising temperature and falling pressure, and decline in frequency with the opposite kind of weather. As the low pressure storm centers approach, nervous irritability and instability increase, and acts of violence multiply. Acute appendicitis shows a similar storm relationship, often coming in real epidemics with rapidly falling pressure.

No. 986

AIR CONDITIONING IN ITS RELATION TO HUMAN WELFARE

By C. A. Mills,* M.D. (*NON-MEMBER*), Cincinnati, O.

As this session of your 40th Annual Meeting is devoted to the health aspects of work in heating and ventilating engineering, I feel hesitant in expressing some ideas of mine, but am led to accept the invitation to do so because of my great interest in the problems involved. For the past six years I have been studying the relation of our climatic and weather environment to health and general welfare, and the results of these studies have convinced me that here is a factor of very great importance in our lives—one, too, which has been badly neglected in the scientific advance of the past 50 years along public health lines.

Engineers have really accomplished wonders in the utilization of inanimate force, making almost anything possible by the mere turn of a switch. You have brought our indoor environment almost completely under control, except for changes in barometric pressure and ionization. Even these will, I believe, soon be conquered by your ingenuity. But how about the other side of the picture—with everything physical within our grasp, do we know just what we want, or what is best for us? Unfortunately medical and physiological research has sadly neglected this field in past decades. There is, however, now taking place a quickening of interest along this line, together with a realization of the tremendously important place climatic environment has in our existence.

Let me briefly summarize for you a few of the known facts about man's dependence on his atmospheric environment. Under carefully controlled laboratory conditions, it has been found that animals adapted for a few weeks to constant moist heat lose in large measure their heat producing capacity and bodily vigor. When subjected to chilling emergencies, they cannot quickly increase their heat production, and so suffer a fall in body temperature and are prostrated. Hand in hand with this sluggish heat and energy metabolism, they show a lowered resistance to infection. Pneumonia, tuberculosis, and other types of infection attack them with greater ease than is seen in more vigorous animals. Under the greater stimulation of a cold environment, animals are more lusty, lay on more fat, are better able to resist sudden chilling and are more resistant to all forms of infections. Most active of all are animals shifted daily from heat to cold, so that they are required to make

* Professor of Experimental Medicine. University of Cincinnati.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934.

frequent and sudden adjustments in their heat production. Such animals stand chilling best of all, but are most sensitive to high temperatures.

These differences induced in laboratory animals by artificial climates exactly parallel differences observed in man under varying climatic conditions. In our cool stormy regions of the Temperate Zone man lives on a high energy plane. He is vigorous, full of pep and vitality, and must be always doing something. In the Tropics and Orient, and in our own Gulf states, there is much less energy available, so that a much greater part of the daily supply must go into the business of mere existence, leaving little for racial or economic advancement. Southerners coming north show their sluggish heat metabolism by the

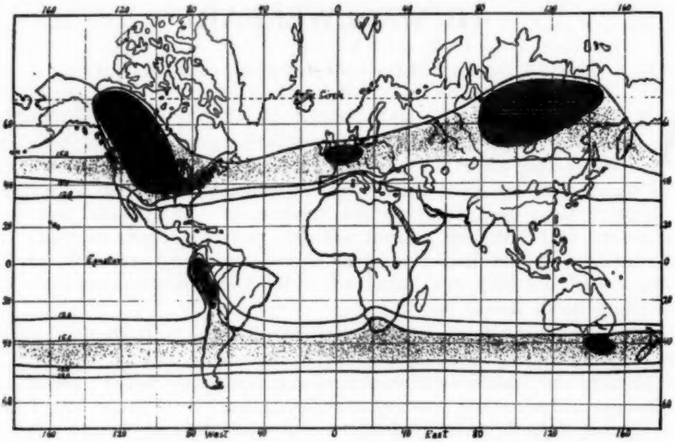


FIG. 1. WORLD MAP SHOWING INTENSITY OF CLIMATIC DRIVE

way they chill under conditions that natives call comfortable. For indoor comfort they demand a temperature 4-8 F higher than we need.

I have constructed a world map based on the intensity of climatic drive imposed on man in the different regions. This is not the place to present the details of calculations used in deriving the indices of climatic stimulation. Suffice it to say that the map, Fig. 1, is based on the best available knowledge we have as to the effects of this climatic drive on man and animals, and represents in a fairly accurate way comparative differences over the earth. It will be seen that we here in north central North America have imposed upon us the most vigorous climatic drive man is anywhere called upon to endure. Perhaps this in a measure explains the astonishing rate of development of the physical resources of the continent. Perhaps it also explains our restless and impetuous zeal for action, as well as our irritated, nervous state. In our Gulf States the climatic drive is very much less, similar to that of the Mediterranean countries of Europe and of Japan and North China.

We here in the North have one rather serious situation to face in regard to this climatic vigor that gives us our abundant energy. Unfortunately we are not capable of unlimited response to this stimulation, and are showing

definite signs of breaking under the strain. Certain of our glands of internal secretion are closely bound up in our energy metabolism and when these fail, disease appears. We derive most of our heat and energy from the burning of glucose, and in this burning the pancreas plays an essential and important part. Thus diabetes is only a sign of pancreatic failure to meet the level of activity demanded. Toxic goiter also represents failure of successful adaptation to stimulation, and we might well class arteriosclerosis and high blood pressure in the same category. The accompanying maps, Figs. 2-5, show how

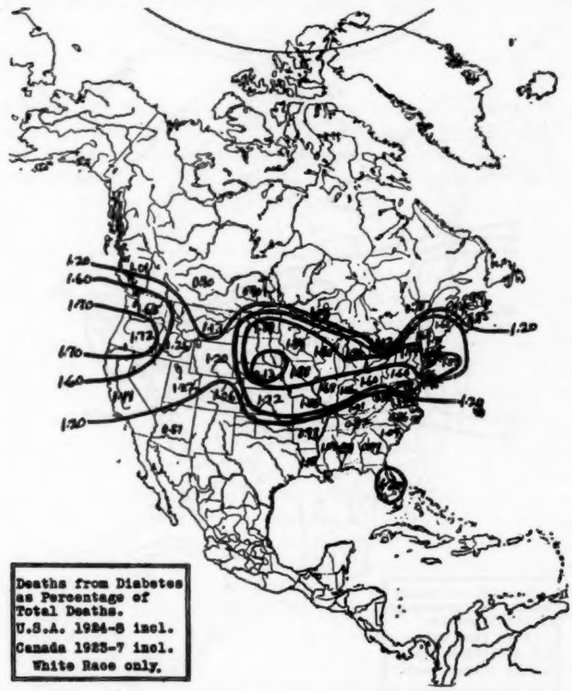


FIG. 2. DEATHS FROM DIABETES AS PERCENTAGE OF TOTAL DEATHS

U. S. A. 1924-8 inclusive—Canada 1923-7 inclusive—White Race only

closely these bodily disturbances follow the climatic drive. Diabetes in the North is a very much more severe disease than it is in the South, reaching its greatest severity in our west central states where the climatic stimulation is most intense. The same holds true for exophthalmic goiter, for pernicious anemia, for arteriosclerosis and a number of lesser metabolic diseases, and curiously enough for cancer.

These advancing signs of bodily breakdown in the most stimulating areas are more than matched by the mental picture. Suicides, mental breakdown,

and nervous disorders are nowhere so frequent as here and in the most stimulating European regions. These bodily and mental diseases mentioned here are the ones causing greatest concern to medical and public health authorities of today. It is essential, though, that they recognize the basic role played by climate if there is ever to be intelligent handling of the situation.

We see, then, that the South profits by its lessened climatic drive in maintaining greater bodily and mental stability. The easier, more relaxed and care-free existence that goes with the lower energy level less often brings on the

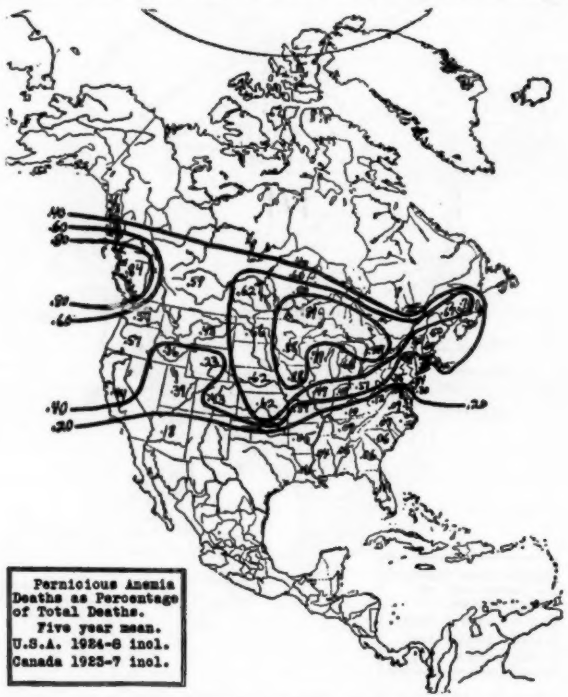


FIG. 3. PERNICIOUS ANEMIA DEATHS AS PERCENTAGE OF TOTAL DEATHS

Five year mean—U. S. A. 1924-8 inclusive—Canada 1923-7 inclusive

diseases of exhaustion. In many ways the southerner is to be envied. With his lower energy state, however, he is more susceptible to infections and shows a higher death rate from tuberculosis, acute nephritis and acute appendicitis. The accompanying map of acute nephritis (Fig. 5) death rates in North America shows just the reverse of the diabetes findings (Fig. 2). Where diabetes is most severe, there acute nephritis causes fewest deaths, and in the South, where diabetes is mild, acute nephritis is most severe. These same findings hold true for the countries of Europe also.

In addition to these disease and energy differences that come from the varying intensity of climatic drive, we have marked effects on health in our stormy regions that come as a result of the wide frequent swings in temperature and barometric pressure. Acute appendicitis attacks and suicides, for instance, have been found to come mainly with the periods of rapidly rising temperature and falling pressure that presage our storms. Colds and pneumonia, on the other hand, come with the days of rapidly falling temperature. But in all these cases, the frequency of attacks are determined by the frequency and severity of the storm changes in the weather, week after week. The central portion of North America is certainly not to be envied in these particular health aspects, and they should be considered in calculating the price of our exuberant energy.

Realization of the many important phases of human life that are affected by climate and weather changes naturally brings up the question,—what are we to do about it? If, in the very stimulating areas of the earth, as in our middle western states, the human machine shows evidences of breaking under the strain, shall we not seek means of lessening the driving force to within safe limits? And in the tropics, where man is held down to a mere existence energy level by the climatic stagnation, should we not attempt to relieve him from the pall of physical depression that is holding him down? If there be a destiny for mankind to fulfil here on earth, then release from the devitalizing effects of tropical and oriental moist heat would enable those hundreds of millions of people to attain that destiny more quickly and certainly. Should not the people of our own Gulf States be given the chance at an energy level that would permit them a more fair competition with the energetic northerners?

But when we ask ourselves these questions, we should also keep in mind certain others that come as natural corollaries. Recognizing excessive climatic drive as the principal basis of our American restlessness and inability to relax, are we justified in foisting upon less energetic peoples this urge to action which will largely destroy their present calm and carefree existence? For there is little doubt but that raising the physical energy level entails always a more irritable nervous system, with less complacency and contentment. It does seem, however, that the very energetic level of life here in America makes us assume that such a state would be best for the whole world. Hence out go our commercial and educational, as well as religious, missionaries to all the backward corners of the earth. The people of China, Patagonia and Greenland shall not live on without modern plumbing and automobiles if we can help it.

Probably most important of all for us here in America is the human stress that comes from our severe and frequent storm changes. The wide swings in temperature and barometric pressure which we must endure every few days do us untold harm. True enough it is these changes that generate in us our high level of energy, but they also bring us to grief and bodily misery. Such diseases as pneumonia, acute appendicitis, *colds,* sinusitis, and chronic arthritis —these all are closely related to our severe storm changes and do much to reduce our health and efficiency. Effects on the nervous system are perhaps even more important, for we must be constantly shifting from the irritation and feeling of futility that comes with the low pressure-rising temperature days, over to the exuberant exhilaration of the cool high pressure periods. We are

never left in a state of calm for more than a few days, but must be always jumping about mentally, due to the effect upon us of these weather changes. It is no wonder we are notional and take up every fad that comes along. Sub-conscious realization of this irritated mental state leads us into the passage of innumerable laws and regulations, most of which we disregard as soon as they are passed. It was felt, for instance, that legal prohibition of alcoholic beverages would do away with most of our wild life in society, but quite the

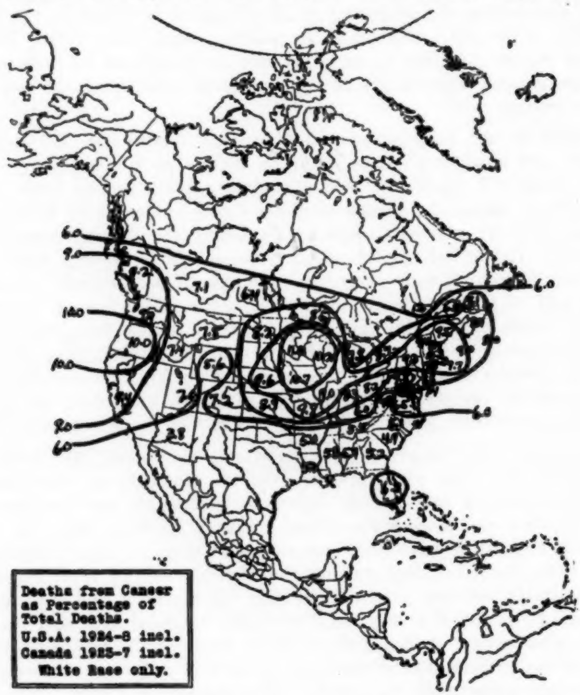


FIG. 4. DEATHS FROM CANCER AS PERCENTAGE OF TOTAL DEATHS
U. S. A. 1924-8 inclusive—Canada 1923-7 inclusive—White Race only

opposite occurred; and now, with the return of beer, a feeling of contentment seems to have settled over the nation.

One cannot stress too much the deleterious mental effects of our frequent weather changes. Continual confusion and changing from depression to exhilaration exhausts us emotionally and leads to an existence full of irritation and restlessness. That is why people from the stormy area find Hawaii, Southern California and Florida such desirable places for rest and relaxation. Particularly do we need such climates as these during our late winter and early spring months, when the northern storminess is greatest and many of us are near physical exhaustion. In our recent years of prosperity it began to look as

though there would develop a migratory movement in the population somewhat similar to the seasonal flight of many birds, and perhaps such seasonal migration will make further headway in years to come. There will, however, always be many millions who cannot get away from home and must remain blindly submissive to the vagaries of the weather unless artificial control is made available for them.

Our greatest hope of climatic control, outside of seasonal migration, lies in

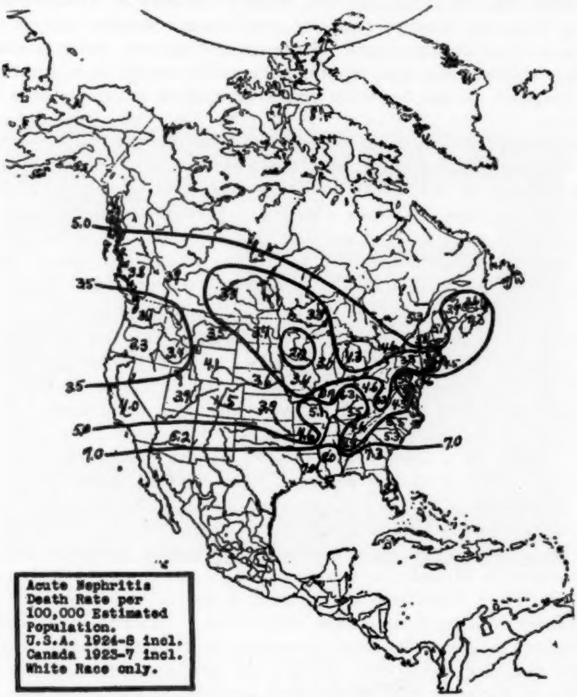


FIG. 5. ACUTE NEPHRITIS DEATH RATE PER 100,000 ESTIMATED POPULATION

U. S. A. 1924-8 inclusive—Canada 1923-7 inclusive—White Race only

the new and rapidly developing industry, air conditioning for human comfort. Through it, and the newer methods of construction that give almost complete insulation from the outdoor environment, we are now able to exercise almost complete control over temperature, humidity and suspended dirt particles in our homes, offices, factories, and public buildings. The cost of the necessary equipment for installation in old buildings is still beyond the reach of many, but installation in properly planned new structures is in the long run more economical than our old haphazard methods of heating and ventilating. It has been shown, for instance, by hospital architects that hospitals, fully adequate

for every reasonable need, can be built and equipped for complete temperature control the year round for not over $2,500 per bed, whereas past costs have ranged from $4,000 to $10,000 per bed, with no provision for cleaning the air or for summer cooling.

It has been pointed out that not only do the heat waves of summer bring on a variety of acute ailments, such as acute appendicitis and summer diarrhea, but that these patients do most poorly when compelled to lie in hot hospital rooms. Surgeons are aware of this lessened vitality of patients during the periods of heat and tend to postpone until cooler weather major operations that are not of an urgent nature. Most often, however, delay is not possible, as in acute appendicitis, and on this account it would seem imperative that surgical hospitals should be fully protected against excessive heat, for both the operation and the after care. It is quite likely that convalescence from operations and fever ailments could be considerably shortened by providing the proper mean temperature level and proper daily range for each day to give the right degree of stimulation. At present, throughout the Tropics and Orient, recovery from a serious illness or operation is much slower than in more stimulating regions. Let us hope that it is only a matter of time until hospitals and the medical profession will be brought to greater use of air conditioning methods as direct aids in securing recovery from illness.

In lieu of expensive equipment in homes, or of seasonal migration to counteract climatic effects, it would seem advisable that there be made available in each city artificial climates of various types, in hotels, hospitals or sanataria, where the patients can go and enjoy the relaxing effects of Florida or Hawaii for a couple of weeks and still be near business or professional interests. Two weeks' rest in bed, in a room kept at 85 F, with 70 per cent relative humidity, would go a long ways toward quieting down the over-dynamic hypertensive individual of our northern cities, and no doubt great use would be made of such facilities when people came to a clearer realization of whither they are heading in our area of greatest storminess and most intensive climatic drive. The same type of air conditioning equipment could be used, also, to produce changeable stimulating environments of any intensity desired in speeding recovery from run-down states, or for use in relieving hay fever and asthma sufferers during their periods of distress. Tuberculosis hospitals could, in any location, be given the same advantageous climatic effects as are offered by the best mountain sanatoria. In fact we can have only a faint realization at present as to the great possibilities such artificial climates may some day offer in promoting human health and function. The field is yet too new and untried to allow of a close evaluation of its possibilities. Its potentialities, however, are alluring.

Another great future possibility is the use of cooling devices in the warmer countries to step up human energy and efficiency to a level more nearly equal to that of the cooler temperate regions. Animal experiments indicate that this can be readily accomplished by just a few hours of cooling each day, such as might be most economically obtained by chilling the sleeping quarters. Everyone is aware of the refreshing effect of a cool night after a hot day, and how the day's heat does not depress nearly so much when a good night's sleep is made possible. It would be a most interesting experiment in mass effects to try such night cooling of a sample population for a few summers to see if their

bodily energy and vigor responded as did that of the laboratory animals. Research engineers are even now working to perfect a less expensive type of cooling device that might bring such an experiment within the means of the people. It must be remembered, however, that the blessings of this higher energy level are not entirely unmitigated. The increased efficiency and vigor carry with them a more restless and irritative existence—a loss of the southern complacency that makes life so pleasant and carefree. Also such a change would render the population more sensitive to the summer moist heat, as well as probably increasing their tendency to diabetes, arteriosclerosis and the like. The American of today, however, usually disregards these dangers, which to him are vague and probably aimed at somebody else anyhow, and grasps at any thing that promises greater wealth. So it is likely that eventual use will be

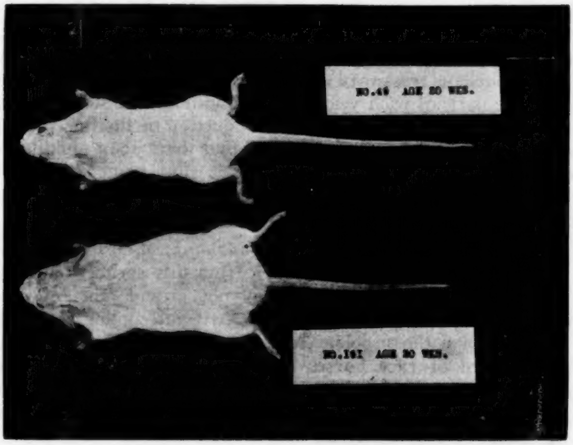


FIG. 6. DEVELOPMENT OF MICE IN HOT AND COLD ENVIRONMENT AT AGE 20 WEEKS

made of the possibilities here outlined for increasing the productiveness of man in the less stimulating climates so that he may compete on more even terms with his more energetic brothers.

Another important aspect of the use of artificial climates is that concerning the breeding and care of domestic animals. Experimental studies have shown that animals raised under optimal climatic conditions are more fertile, mature earlier and are in all ways more vigorous than are those subjected to moist heat. With laboratory mice the differences in fertility and growth are marked. Litters are usually almost twice as large in the cold as in the hot environment, and in addition they are much more lusty. With the small litters of the hot room, a large number are born dead or die soon after birth. In addition, although the animals mate freely in either heat or cold, conception is almost invariable in the cold but difficult to attain in the heat. These findings, if applied to stock breeding would mean a very great handicap for the regions that have prolonged periods of moist heat such as is met with in the Gulf states.

Just as important as the fertility differences, however, are the changes in rate and type of growth produced by the different environments. By proper stimulation, growth is at all times far ahead of that seen under moist heat. The animals raised in the cold are more plump, their bodies are larger, and their weight markedly higher. The accompanying photograph, Fig. 6, of representative mice from the two groups brings out this point very well. The total body length, including the tail, is always greater in the hot environment, but the marked difference in body size is striking. These facts should be of great value in the raising of stock for meat, whether it be beef, pork, mutton or fowl.

The few fortunate areas of the earth where meat supplies are most easily and cheaply produced are also those characterized by a stimulating climate. The States of our North and Western Plains, Australia, the Argentine, Central Europe, the Mexican plateau—these all are favored by a climate that induces great vigor of body, rapid growth and great fertility.

The Tropics and Orient, on the other hand, and also the states of the Old South (from Louisiana eastward) are held down by devitalizing moist heat for all, or a large part, of each year, and so find domestic stock raising difficult. Breeders have tried, time after time, the introduction of the fine vigorous strains from the more stimulating regions, only to find them soon sinking to the lean stringy type so common in those regions. It is almost impossible to keep up a herd of good milk cows there, principally because of the toll taken by tuberculosis and the climatic suppression of metabolism.

Is it not time that the knowledge now available be used to liberate the South (and all regions afflicted with moist heat) from this serious handicap to their economic development? Why should they not provide their own supply of good meat, eggs and milk in abundance? Artificial cooling, properly applied, would I am sure solve the problem for them. Dog raising, poultry growing, and the proper care of race horses demands just as much consideration of these climatic effects and the means of their relief. These few remarks will serve to give you some insight into the possible future expansion of the air conditioning industry for both man and animal shelter.

Whether we will eventually be able to escape the disturbing effects of our wide swings of temperature and barometric pressure as the major storms pass by is questionable. The more time we spend each day in an environment artificially regulated, the nearer could we come to this goal. But with our goings and comings, together with our modern ideas on fresh air while we sleep, we are exposed to the uncontrolled vagaries of the weather for at least half of winter, and to far more than half during the rest of the year. Are we headed for a more rat-like burrowing existence indoors as the decades pass, or by natural selection are we to permit the climatic drive to eliminate those who cannot adapt to its severity? The latter is a matter of centuries and beyond our control, while the former is even now in actual process of accomplishment, with rapid strides being made in achieving independence of the external weather changes.

The future likely holds for us some striking changes in our idea as to the value of fresh air. I have little doubt that another decade will see great care being exercised to avoid excessive climatic stimulation during our cooler seasons, and one of the first fads to go will be the sleeping with wide open windows. The day is surely coming when we will intelligently seek to smooth

off the severe weather changes here in the North, and to raise the energy level in the South by night time cooling through the warm season. Our aim should be to relieve man everywhere of the handicaps imposed by climate, by securing a universal mean best suited to our bodily capacity.

DISCUSSION

MARGARET INGELS: I would like to know whether, in the correlation of the data where outdoor climate conditions were plotted against diseases, any correction was made for the communities where people are living within shelters? For instance, it is very easy to conceive of New York people not being exposed to outdoor weather conditions as much as the people who live in North Dakota. Has there been any correction made for this tendency of city people to carry on all activities within shelters, especially in the cold climates?

THORNTON LEWIS: It has been a long time since we have had a paper as stimulating to thought and further study as this paper of Dr. Mills. I think the paper is very valuable as a means of bringing out many points which should interest us as engineers and some of us as scientists.

C. P. YAGLOU: I read Dr. Mills' articles some time ago and found them particularly interesting. I was surprised to find out how closely the results of Professor Ellsworth Huntington and Dr. Mills agreed. No matter what you take, whether it is death rate, suicide, appendicitis, or human conception rate, or whether you take death rate of infants less than one year of age, or premature infants, you will find that in the long run the optimum climatic conditions are about the same for all groups. Some people won't believe it, but time will tell.

As Dr. Mills pointed out, we really do not know all the climatic factors involved. We do not know the exact effects of temperature, humidity, changes in temperature, or ionization, but so far as we know at present, temperature appears to be the strongest factor or at least a strong agent that may be associated with other unknown factors.

C.-E. A. WINSLOW: I should like to study more thoroughly, some of the statistics of the diseases in different parts of the country. It is an extremely complicated question. For instance, in the case of goiter one has to consider the specific distance from the ocean, in relation to the amount of iodine present in the natural waters. Diseases like cancer and diabetes are so closely related to the economic status and sugar consumption that that introduces possible factors. There are a great many things one would have to study in order to make up one's mind as to the harmful effects of over-stimulation.

On the other hand, I am entirely convinced from Dr. Mills' work and from Professor Ellsworth Huntington's, which I have followed for a great many years, of the broad essential relationships between civilization and a cool and stimulating climate. I think that is proved beyond any question.

These long-range figures that Dr. Mills quoted from the Weather Bureau are of extraordinary interest although very disheartening. It is evident that we are falling back into the period of the Dark Ages, but we may hope that at least those areas in the world which have a stimulating climate, even in a hot period, may manage to survive and start the next wave of civilization.

DR. W. J. MCCONNELL[1]: Dr. Mills has given us a great deal of information and considerable material for thought, particularly in regard to diseases. I do hope, however, that Dr. Mills' paper will have some influence on educating our indoor workers to appreciate a more stimulating temperature. We find that the office build-

[1] Assistant Medical Director, Metropolitan Life Insurance Co.

ings in New York City which are not humidified require a temperature of around 80 deg for comfort. In our one air conditioned building, we dare not lower the temperature below 73 deg, although we maintain a relative humidity of between 40 and 50 per cent. When the temperature is under 73 we have a number of complaints. So, if Dr. Mills' work will indicate the desirability of lower temperatures indoors, it is going to be of tremendous value.

C. S. LEOPOLD: Dr. Mills' paper dealt largely with outdoor climate and relatively long periods of change. I have recently been questioning whether or not a similar effect can be demonstrated in our indoor climate where people are exposed for periods in excess of 2 hours.

The summer optimum line on the comfort charts shows the zone of 60 per cent comfort for 98 per cent of individuals. I do not know of any results today that show 98 per cent comfort to any one individual for prolonged indoor exposure. This raises the doubt as to whether there is any constant optimum wet bulb and dry bulb for human comfort or well being and further suggests the advisability of experiments to indicate whether or not a better feeling of comfort could be obtained by keeping a relatively constant effective temperature, but continuously varying from warm dry to cool moist. This would result in a form of physiological exercise in that the body employs different mechanism for eliminating latent and sensible heat. There is some reason to believe that the desired stimulation can thus be obtained with substantially constant comfort. This procedure differs from the older experiments in which the dry bulb was varied without compensatory changes in the wet bulb, with consequent variations in the effective temperature and, therefore, comfort. These effects may be desirable from the standpoint of stimulation, but are hardly desirable in public meeting places, offices, or even homes, where comfort is the first requisite.

A. A. ADLER: The question of the most desirable temperature and relative humidity to be maintained in a given space involves many factors as yet unknown. The comfort chart as given in THE GUIDE states certain relations under given conditions. However, there are cases peculiar to hospital operation which cannot be levelled by any uniform rule. This raises the question as to the viewpoint which should prevail in such cases. It is different from the viewpoint of the patient at low vitality, the operating and assisting surgeons, nurses and sometimes medical students. The mental state, manner of dress, previous exposure, etc., all have their effect on the individual's attitude in any uniformly conditioned space. Accordingly fundamental research should have for its object, among other things, the nature of the problem under investigation.

To this end, we should know how energy is produced in the body, the transformations that go on within, the mechanism of energy utilization and the dissipation of the surplus heat. By means of a heat balance and the various limitations of energy generation and dissipation, we may be in a position to do orderly research. In addition we may be in a position to advise equipment manufacturers as to the nature of the problem before them. In this direction Dr. Mills certainly takes a courageous step in collecting fundamental facts. An investigator charged with the assimilation of much of the existing data on the physiological side of the subject is confronted with the chaotic state of the subject and will perhaps conclude that much of the opinions previously expressed would better have been left unsaid. There is a wide difference between physical and physiological investigations with respect to the reliability of the data collected. In the natural sciences we always have the opportunity to appeal to experiment and the results are always consistent under similar conditions. The exact conditions which exist in physiological experiments are usually difficult to duplicate and hence conclusions hastily drawn are apt to be misleading.

W. L. FLEISHER: I haven't seen any results that would confirm Dr. Mills' paper on disease, but I have seen some very interesting results that have been tabulated

on the production of work under different conditions. In one very large industrial plant the relative humidity was a factor and was maintained at about 74 or 76 per cent. The plant was equipped with air conditioning and refrigeration apparatus. Before proceeding with their new plants, which would have required about the same type of climate, they made a test of the production of the operators under cool conditions and under the normal conditions that would be obtained in the summer. This was in one of the southern states. Although those results have never been published, I saw information extending over 2-day periods in which between 12 and 18 per cent more goods were produced under artificial cool conditions than under normal conditions, showing the increased productiveness of human beings under favorable conditions. I imagine there have been other tests or other statistics of that kind not given out by industrial concerns. The temperature ran as high as 95 deg outside and it was maintained at about 76 deg dry-bulb and 75 per cent relative humidity under artificial conditioning.

DR. LEONARD GREENBURG: I feel that the point which Miss Ingels mentioned bears repetition, and for this reason I should like to bring to the attention of the audience the fact that the temperatures which Dr. Mills has placed on the screen, and those of Dr. Ellsworth Huntington, whose work is along these same lines, are average outdoor temperatures for the 24-hour period and they should accordingly be interpreted on this basis. I do not believe that Dr. Mills intends us to assume that these average temperatures are the optimum ones for indoor conditions.

MR. FLEISHER: Some years ago a much debated subject was the question of open-window ventilation. I remember a rather heated discussion in Buffalo. I made a suggestion that one of the most unfavorable things in our modern mechanical ventilation was the monotony of conditions and at that time there was very little response. I stated that probably one of the advantages that could be cited for open-window ventilation was that it was so uneven. I got very little response because it didn't accord very well with the very careful control of temperatures and humidities that we were all striving for. But I think that would be an interesting experiment in the maintaining of health, a variation in conditions from rather too warm to too cool and back through the cycle over a definite period. It is one of the things that I have never seen attempted, although I have suggested it at least ten or twelve times to this Society. All of our work has been carefully planned to give absolutely uniform conditions without variations, whereas my opinion and theory has always been that probably it is variation through a mean condition that gives the most comfortable conditions.

DR. C. A. MILLS: The question brought up by Miss Ingels regarding the correlation of human statistics with outside weather data, when most city people are exposed to such weather for only a small part of each day, is well taken and does serve to cast doubt on urban statistics. My data, however, apply equally well to predominantly farming populations. Greatest significance is derived, though, from results on experimental animals under carefully controlled conditions, where outside weather conditions were imitated and results obtained that fitted in well with human findings. We had, for instance, continuous moist warmth, continuous cold, and a daily shifting from warmth to cold to mimic our storm changes. Daily cooling of about 25 F for 6-8 hours seems to produce just such a stimulation and step up in heat and energy production in animals as is brought about in people by the outdoor winter weather changes of our northern states.

In real life (even in cities) we have just this type of stimulation. For 8-10 hours a day we work in rooms at 70-75 F, while on the way to and from work we are exposed for an hour or two to outdoor cold. Through the evening we are again warm, but finally we retire in a bedroom freely exposed to outdoor cold through wide open windows, and breathe the cold air 8 hours each night. As I have pointed out, the degree of stimulation under which people of the northern storm area are living

seems already too excessive for human welfare. Bodily and mental breakdown is becoming more and more frequent, and is coming ever earlier in life. We need, therefore, to seek means of easing off the drive on the population in every way possible. And I believe a greater degree of control over night-time temperatures is highly desirable. Probably a 25 F fall in temperature from the daytime level will be found best for sleeping quarters. In the light of recent findings, the 40-80 F drops we have been accustomed to experiencing in our sleeping quarters no longer seem sensible.

I am well aware of the inaccuracies and possible fallacies of human vital statistics mentioned by Dr. Winslow. So far, however, they represent our only source of information on human response, and, if used with proper caution, may yield valuable findings. Particularly gratifying has been the close way in which our animal findings have corroborated and supported our human findings.

In my studies on human diseases I did not mean to infer that the climatic drive itself causes diabetes or toxic goiter, but that the stress imposed on the body pushes an increasing percentage of the population too near the point of metabolic exhaustion, and it is from this exhausted fraction of the population that the ranks of the diabetics and goiter and pernicious anemia patients are recruited.

Some one brought up the question of the Eskimo and the stimulation he must undergo. The truth of the matter is that he lives under conditions of tropical warmth with temperatures around 85 F in his igloo most of the time. During his infrequent forays he is bundled in warm furs and gets stimulation only through breathing the cold air.

Europeans keep indoor temperatures fully 10 F lower than we do, and hence have less of a contrast between indoor and outdoor conditions. Nor do they insist on such free exposure to cold night air as we have been wont to consider essential to health. I believe the time is coming when we will give up our fresh air fad in favor of continuously controlled conditions.

There are several points that we need to consider and study in regard to climate and weather effect. I should like to see a greatly intensified study in several places over the country, in several laboratories, of climatic effects and what should be done about them. The medical profession certainly is not yet ready or able to specify the conditions that would be best for man in a given region. Your air conditioning industry is far ahead of medical knowledge in this field.

Dr. Yaglou spoke about ionization. I said in discussing his paper that I believe there is a great importance to us in these cold, heavy air masses that come along. I think ionization will be of great significance when we know just what that significance is, but I still believe that temperature change and level are perhaps the major factors.

Now, as to the increased efficiency in the southerner. You cannot expect to get any great step-up in efficiency of a southerner by cooling him suddenly for a short time only. If you chill animals adapted to the hotroom, their body temperature falls quite readily. So it is with these people in the industrial plant; it takes time for them to respond. The northerner will respond quickly to temperature changes; the southerner will not. What he needs is to be chilled some each day throughout the warm period for a year or two. Then you will really see a step-up in his mental and body efficiency.

In regard to the varying indoor conditions, whether we should have constancy throughout the 8-hour work day or whether we should vary that period, I am incompetent to say. It may matter for comfort, but I do not believe it would matter on an energy basis. It might have an effect on efficiency in the finer details of technical work, but stimulating an individual once every 24 hours gives the maximum response in energy. If you stimulate him three times a day, it is no better. In fact, you can exhaust him.

No. 987

ENGINEERING IN THE HOSPITAL OF TOMORROW

By CHARLES F. NEERGAARD * (*NON-MEMBER*), NEW YORK, N. Y.

AT a recent meeting of the New York Chapter of this Society the writer presented a paper which ventured to challenge certain commonly accepted principles of hospital engineering. Consequently the critic has been invited to turn creator and formulate the type of plant the hospital should have and what it should accomplish.

There are many chapters in the history of hospital engineering which are chapters of extravagance and error. The hospital's stated requirements have often been contradictory and many of its mechanical standards built up on fallacious theories, rather than on scientific analyses of precise needs. While the hospital is a laboratory for medical research it should not be the subject of mechanical experimentation. Its mechanical plant has become so elaborate and complex that the cost of installation, operation and maintenance is oftentimes indefensible. Yet new refinements and luxuries are constantly being urged. The ideal plant is one where each feature is designed with calculated simplicity to do its job adequately, regularly and economically.

FINANCIAL PROBLEMS

The major problem in the hospital world today is the financial problem. During the depression demands for free care have fast multiplied while earnings and contributions have fallen to an alarming extent. It is the theory of the French savant, Abbé Le Maitre, that the universe is expanding and collapsing at the same time. With the hospital this is a condition, not a theory. There is many a tragedy in excessive but irreducible cost of plant operation, where every dollar wasted in the power house means a dollar less for the care of the patient.

The mechanical plant holds an important place in the financial picture. On the basis of average experience a 100 bed general hospital in the New York area built in 1930 would have required an investment for building and fixed equipment of some $650,000, of which about $200,000 would be represented by power, heat, plumbing, ventilating, elevators, sterilizers, kitchen and laundry

* Hospital Consultant.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

equipment and electrical services. In 1932 the hospital would have spent $160,000 to operate in which would be included only the actual cost of replacements and repairs, but a small percentage of the inevitable reckoning for depreciation and obsolescence. In the figures of 30 New York City hospitals for 1932 there was a spread of from 3c to 9c of the expense dollar for fuel, power and light alone. Thus the 100 bed hospital, if it had an economical plant, might spend as little as $4,800 for these essentials or if handicapped with an extravagant one, as much as $14,400.

Whatever may be said of the qualifications of the operating engineer, scientific planning and sound judgment can build into a plant operating economies which will be constant and largely inescapable. The cost of the hospital structure and the cost of hospital operation have increased to a point where it is imperative that our thoughts be in terms of major economic reforms.

The hospital needs the same promise of salvation as is appearing on the railroad horizon, where revolutionary changes in design and construction are producing streamline passenger trains of one-third the weight which have 50 per cent greater speed at half the operating cost of those they replace.

THE NEW HOSPITAL PLAN

In the light of past experience and present knowledge of planning, construction and equipment, and based on hope, perhaps, rather than prophecy, the hospital of tomorrow is going to be built at from 25 per cent to 40 per cent less cost than the hospital of 1930, given the same labor and material levels. The fixed charges for operation and maintenance, interest and depreciation, must be reduced in even greater ratio.

While it is not within the province of this paper to anticipate architectural evolution, it will be necessary to blaze the way in certain particulars, to establish the requirements for the mechanical plant. The 100 bed hospital structure will be smaller than its 1930 prototype, lighter, more compact, quieter, longer lived, and with a better insulated envelope. The mechanical services, simpler and more efficient, will provide better conditions for patients and those who care for them.

The building will be smaller through the elimination of waste space and waste cubage and because of fewer beds needed in reserve. The supreme quality of elasticity in the plan will make it possible for a hospital with 100 beds to meet the demands of a district where an average of 85 patients requires bed care, for which in the past 130 beds have been provided.

The building will be lighter, fireproof with a welded steel frame, the steel tonnage being radically reduced by the use in walls, partitions and floor slabs of light cellular concrete. This weighs from 40 to 70 lb per cubic foot depending on requirements, yet has ample strength.

Insulation

In addition to its reduction of the dead load, cellular concrete *for walls* presents the advantage of lower heat conductance and higher specific heat than most other insulating materials: *for partitions,* effective insulation against sound transmission in minimum space, a 4-in. block plastered both sides having

a coefficient of some 38 decibels. Thus a *sound proof* partition of 5½ in. replaces one of 8½-in. thickness; *for floor slabs,* poured in 4 ft spans, it provides effective sound insulation between the floors. The cinder fil[illegible] be omitted. This in the past has served to cover irregularities in pouring [illegible]rches and to conceal the pipes and conduits no longer to be run in the flo[illegible]ture. The usual separate screed coat will be unnecessary as the expansion of this concrete when poured can be controlled within a quarter of an inch and the floor slab itself, in a single operation, rolled and finished to a smooth level surface, ready to receive the finished flooring.

The building will be more compact. Scientific planning will apply the principles of industrial efficiency in every department to minimize waste space, waste motion and waste effort.

The building will be quieter. Sound insulating partitions and the liberal use of acoustical treatment will confine and absorb internal noises at the source. Quiet operation will be a prerequisite for every piece of equipment and all engineering specifications will contain a covering clause binding on the manufacturer and general contractor that noise is a defect which will lead to rejection. All noise is relative and a fair acceptance test difficult to formulate without assurance of adequately insulated bases and acoustical treatment.

The building will have a better insulated envelope. Buildings today are structurally wrong when air leaks through the walls like sieves. Infiltration of 30 to 60 cu ft of air per minute around windows is not exceptional. Now with the steel frame a curtain wall replaces the bearing wall; thin ashlar simulates massive masonry; thermal insulation is conspicuous by its absence. As it carries no load, the ultimate wall may be a thin wall, insuring maximum hours of precious sunlight in each room. It will be a wind- and weather-proofed overcoat with an inside lining which will feel warm and hold the building's body heat.

Some day the entire exterior may be faced with aluminum, copper or stainless steel, reflecting 95 per cent of the sun's heat. But for the present it will be ncessary to be content with conventional brick or stone veneer effectively waterproofed with a backing of cellular concrete block and an inch or so of insulating material of low thermal conductivity. Whatever insulation is used,—rock wool, metal foil, rock cork, cork board, gassed rubber or some material as yet in the laboratory stage,—the conductivity of the wall should be less than half that of the present standard masonry construction. In the roof equally effective insulating measures will be used.

Eventually the ideal wall will be designed. It should heat up quickly, in minutes, not hours, after the window is closed on a winter morning, and feel warm all day, for warm wall surfaces are fully as important to health and comfort as proper warmth and moisture in the air. There will be no plaster. The walls and ceilings will be of panels finished with synthetic resin of whatever color desired, resistant, permanent, washable, and the painting problem will cease to be.

Windows

The pioneer studies of Voorhees and Meyer on Window Leakage and later reports from the A.S.H.V.E. Research Laboratory and the cooperative work

Table 1. Estimate of the Installation and Operating Cost of Heat and Ventilation in a Hospital with and without General Air Conditioning Based on the Plant in an Office Building with 3,000,000 Cu. Ft. of Construction and Floor Areas Sufficient to House a 300 Bed Hospital

The office building is in use 8 hours a day, 5½ days a week, the hospital 24 hours a day, every day of the year.

	Office Building	Hospital without Air Conditioning	Hospital Equipped with Air Conditioning
Installation Cost			
Heating Plant	$ 68,000	$68,000	$ 68,000
Air Conditioning	165,000	......	150,000
Exhaust Ventilation	7,500	20,000	7,500
Total Investment	$240,700	$88,000	$225,500
Operating Cost Omitting Labor			
Fuel and Heating Supplies	$ 6,100	$15,000	$ 15,000
Power and Maintenance Air Conditioning	17,000	......	25,000
Ventilating	1,000	4,000	2,000
	$24,100	$19,000	$42,000

Table 2. Summary of Steam Requirements

Use	Hp Needed	Steam Pressure Needed	Daily Hours at Full Load
General heating system	90	5 lb	24 hour
Special heating, operating and delivery rooms	3	5	8
Domestic hot water supply	20	5	16
Laundry	15	100	6
Kitchen and dishwashing	10	20	6
Sterilizing	15	40	5
Total	153	..	..

at the University of Wisconsin have demonstrated the extravagance of poor windows. Hospitals have found the solid steel pivoted window highly satisfactory. The sash, closing against felt gaskets, shows negligible leakage. The felt cushions absorb vibrations, exclude street noises, act as shock absorbers in reducing breakage, and show little deterioration after 15 years.

A major problem, as yet unsolved, is a simple and economical control of heat loss through window glass. If double glass can be produced, parallel sheets fused with glass spreaders, forming an air space, filled with a dry gas, *e.g.* carbon dioxide, at atmospheric pressure, there will be a high degree of insulation and no condensation. Research should sooner or later produce a commercial product for which there will be a large market.

With walls and windows as described the radiation in the building may be reduced from one-half to one-third of what is now used and the boilers propor-

tionately. The additional cost of the insulating measures should not exceed the resulting cut in the cost of the plant. The fuel saving will be an annual extra dividend.

AIR CONDITIONING

Perhaps the most popular recommendation which could be made for heating and ventilating the future hospital would be to turn the entire problem over to the air conditioning engineer and let him install a central plant with a robot in the engine room surrounded by registering and recording gauges, and instruments of precision, with which to regulate all of the elements of heat and cold, moisture and clean air throughout the building. Admittedly the hospital of the future must have better air. It must keep pace with the findings of the physiologist and provide whatever will contribute to the safety or cure of the patient, but every item of increased expense must be conclusively vindicated.

The present trend in central air conditioning does not appear to be in conformity with the hospital's pocketbook. An office building erected in 1932, comparable in bulk to a 300 bed hospital, with an air conditioning plant, then conceded to represent the most advanced ideas, will serve as an example. The heating system is entirely separate from the air conditioning. In each room conditioned air is introduced in a horizontal stream through grilles at the top of the inside partition and circulation maintained through exhaust ducts similarly located. Many protesting tenants have defeated the plan that all windows be kept locked. It is evident that complete abandonment of the open window must be deferred until human nature has changed more than a little.

The building has 10 zone fan rooms. There are 15 fan and two pump motors ranging from 3 to 25 hp with a total of over 100 hp. The compressor motor of the 300-ton refrigerating unit is 250 hp. If the building were a hospital, this would mean a ton of refrigeration and more than a horsepower motor capacity for each bed.

Applying the experience of this building to a hospital is illuminating although the hospital figures are more suggestive than accurate, as shown by Table 1.

On this basis the 300 bed hospital would spend $137,500 to install central air conditioning, or 7 per cent more than 1930 construction costs ($6,500 a bed). The $23,000 additional for operating expenses would represent an increase of 80 per cent over the average expenditure in 1932 for fuel, light and power ($96 a bed).[1]

The hospital is in hearty sympathy with the ends and ideals of air conditioning but the present cost levels are beyond its reach. Any other conclusions would be inconsistent with the purpose of this paper which is admittedly to commandeer the ingenuity and skill of the members of this Society and the resources of its research facilities to the end that the hospital may be enabled to house and care for its patients without having to spend so much money. The late Bill Guard's brief definition of the Einstein theory, "There is no hitching post in the universe" is suggestive of how far the engineering profession can be expected to go to achieve so worthwhile an end.

---

[1] Average annual operating cost $1,600 per bed. Fuel, light and power 6 per cent, or $96 a bed.

NEW MECHANICAL PLANT

In suggesting mechanical desiderata for the hospital of tomorrow from the user's standpoint the author does not hesitate to project theory beyond current practice as it will be the engineer's responsibility, not his, to bridge in some effective manner any gaps between present procedures and future ideals. As crystal gazing is essentially foreign to engineering it has been the aim in this structural and mechanical synthesis to forecast only things which have been done, are about to be done, or reasonably should be done.

The premise seems admissible that with the New Deal, the mercury boiler, power pools and hydro-electric expansion the hospital may anticipate cheaper electric power.

The power plant of the past has been a considerable industry in itself, heavily subsidized by the hospital. The operation and care of high pressure boilers, steam mains, pumps, valves, traps, heaters, water treatment, requiring two or three shifts of licensed engineers and firemen, has constituted perhaps the greatest source of worry and uncontrollable expense to the management. But why a high pressure plant? Few hospitals generate electricity and but little high pressure steam is really used. Consider the picture. To supply the usual steam requirements for a 100 bed hospital most engineers have logically installed two 150 hp boilers, one a spare—a few conservatives, two of 100 hp.

The use of equipment in the last three items of Table 2 is exceedingly irregular so that in the absence of actual records the estimated hours of full load can only be approximated. Obviously even a 100 hp boiler would rarely be overloaded except in extremely cold weather, yet 100 lb steam pressure must be maintained a large portion of each day with a waste, particularly in the summer months, which is not cheerful to contemplate.

In the Hospital of Tomorrow the high pressure plant is practically eliminated. Its passing is due partly to its high cost and poor adaptation to hospital needs, but more as a natural consequence of engineering developments which have, and will, make available new and more efficient equipment, accomplishing better, at less cost, what the hospital requires. Much of the equipment will be operated by gas and electricity. With the old system will depart the wastage from overheated buildings, heat losses and leaks in mains, drips, pumps, traps, valves, etc., which with the heavy engine room payroll have contributed so largely to the red ink total.

Hot Water System

The hospital has found hot water heat to be the most comfortable and satisfactory. With a properly designed plant, it will heat a given building, engineers assert, at less cost than any other appropriate system. The new heating will be of the forced hot water type, with water carried at somewhat higher temperature than present practice, to permit of smaller radiators and piping, with a range from a normal of 140 deg to a high of 220 deg on the occasional zero day. Thanks to well-insulated walls, double glass and hotter water, the radiators of a type giving the maximum of radiant heat will be approximately one-half to one-third the size of those now in use. They will be recessed under the windows but not enclosed. As the temperature of the

water and heat levels throughout the different parts of the building will be regulated in the boiler room, all radiator traps and valves (except key shut-off) will be omitted. What this will save may be judged from the fact that in a 100 bed hospital studied, there were 260 radiators with 520 valves and traps. In patients' rooms and wherever warranted there will be a supplementary local control,—a little unit which will be called a "booster"—under the radiator, both concealed by a small removable facia. It will consist of a small fan, a fresh air inlet, removable filters and a damper mixing fresh and recirculated air in desired proportions. It will be similar in principle to the unit ventilator but smaller, simpler and far less expensive. The fan, about 1/20 hp, will be noiseless, dependable, but cheap,—preferably not electric,—perhaps compressed air. The booster, when started, will increase the temperature of the room by speeding up the circulation of air over the radiator, or cool the room by drawing in air from outside. It will run only semi-occasionally when the occupant desires more or less heat than the normal provision, whereas the unit ventilator fan must run at all times to heat the room. Partly to compensate the valve and trap manufacturer for the volume of business which this new scheme will take away from him he may be given the job of making the booster with an appropriation for each unit equivalent to the present price of a good valve and trap. Less than half of the radiators will be equipped with boosters.

The Boiler Room

With the reduced load and decreased radiation instead of two boilers of 150 hp each, one always idle, there will be three identical boilers of 25 hp each, cross connected. Two of these will be used as hot water heaters, the third as a high pressure boiler supplying steam to the laundry and for dishwashing, and acting as a reserve for the heating units. As high pressure steam is used not over 8 or 10 hours a day but one licensed engineer will be needed. The heating units will require cleaning rarely more than once a year. A boiler plant designed according to this principle has been in operation since 1913 and given almost perfect service. Hot water will be generated with full automatic oil or gas firing or by metered steam. The hot water boilers will be separately controlled, one supplying radiators in north rooms, the other those facing south. Thermostats controlling the temperature of the circulated water will be set at a point, depending on outside wind and weather, to produce the temperature desired on the floors. The Superintendent will have a new measure of control over his Engineering Department. By means of distance reading thermometers he can determine the temperature of any floor at any time, without leaving his office. Various automatic recording devices will chart wet and dry bulb temperatures, fuel consumption, etc., giving a continuous picture of what the power plant is delivering and what it is costing.

Hospitals are notoriously overheated. A large hospital recently visited was everywhere uncomfortably hot. One unoccupied private room was like an oven, and the nurse immediately opened both windows wide. Nurses automatically turn off the light, but never the heat, when a room is not in use. No thermometers were to be seen until the air-conditioned operating room was reached. Here were two thermostats labelled with surgeons' plaster "Keep at 75 deg." The thermometers stood at 84 deg. How much does it cost a day or

TABLE 3. COMPARISON OF ELECTRICAL AND STEAM REQUIREMENTS FOR STERILIZING

Apparatus	Size in Inches	No. of Units	Current Consumed per Sterilization		Steam Consumed per Sterilization	
			@ KW	Total KW	@ Pound	Total Pounds
Instrument Sterilizers	19 x 9 x 8	3	2	2	10	30
Instrument Sterilizers	20 x 12 x 10	3	3	9	10	30
Utensil Sterilizer	20 x 16 x 16	7	6	42	15	105
Bedpan Sterilizer	5 pan	6	6	36	15	90
Pasteurizer	36 bottle	1	..	3	..	10
Dressing Sterilizer	20 x 28	1	..	8	..	30
Dressing Sterilizer	20 x 36	1	..	9	..	34
Autoclave Lavatory	16 x 24	1	..	6	..	20
Water Sterilizers, pair	10 gal.	1	..	10	..	50
Water Sterilizers, pair	20 gal.	1	..	20	..	100
Total	....	25	..	149	..	499

Total cost @ 1c per kwhr, $1.49; @ 80c per thousand pound $.39.

a year to overheat a 1,000,000 cu ft building by even 5 deg? This is a problem which the consulting engineer seems constantly to pass on to his hospital client.

It has been shown that a 150 hp boiler is operated at 100 lb pressure although only the laundry and dishwashing, 10 per cent of the total load, actually require high pressure and that for but 6 hours a day. The steam plant consumes fuel 24 hours a day the year round to maintain pressure and make up the constant loss by radiation through piping, regardless of whether steam is being used or not. Gas and electricity will be substituted for steam in kitchen equipment; and in sterilizers, which are only used intermittently, to eliminate expense when they are idle. Irrespective of the rate paid for current it is evident that this will effect major economies.

Such changes as the foreging obviously involve many breaks with tradition but wherein is the scheme not sound?

2 in 1 Plant

The new plant will provide all of the essentials of conditioned air simply and economically. The same pumps, mains and radiators that distribute hot water to heat the building in winter will carry cold water or brine to cool it in summer. A drip pan under each radiator with drain connection will catch the condensation forming on the cold radiator in summer. Humidifying units in the corridors, controlled from the engine room, will circulate mist from the domestic hot water supply which will penetrate throughout the building. It is estimated that in the Middle Atlantic States the average winter temperature is 39 deg and the average outdoor humidity 75 per cent, and that for every 10,000 cu ft of air space in a home 5½ gal of water should be evaporated daily to keep the air healthy and comfortable. Thus the 400,000 cu ft which should be humidified in the hospital would require 220 gal a day. The booster unit

in the patients' rooms will, when needed, supply air motion and fresh filtered air with windows closed and outside noises excluded.

How nearly will this adaptation of the heating system to air conditioning provide the results of a central plant? Cooling can be accurately controlled: dehumidification may at times be less than ideal, but humidification ample.

Cost Comparisons

And what about operating cost, particularly in summer? In the New York District there are on the average some 5,000 hours in the year when heating is needed, and 500 when cooling is desirable, which with the insulated walls proposed will be considerably reduced. The Weather Bureau figures for the summer of 1932 show 485 hours with relative humidity between 75 and 85 per cent and 529 hours above 85 per cent—say 1,000 hours when dehumidification is required. How many winter hours there are when artificially heated air is thirsting for moisture is not on the record.

At the request of the author an engineer associate made a computation of the comparative costs of supplying the same cooling by the two systems in the hypothetical 100 bed hospital. Assuming that 400,000 cu ft of air is to be cooled, the volume of brine circulated at 45 to 50 deg would be 40 gal per minute or 99 tons for a 10-hour period. This would require a 15 hp compressor motor. Using the fans and ducts with the same formulae as applied in the office building and allowing three air changes per hour for that portion of the total air recirculated would mean 17,600 cfm or 360 tons of air for a 10-hour period. Motors aggregating 70 hp would be required. To the lay mind these figures prove little but suggest much. It would appear that the hospital might content itself with the smaller tonnage.

It does not seem unreasonable to challenge the engineer to build, around the principles suggested, a plant which will maintain the hospital's air conditions close to the borders of the comfort zone and give it something which, if not perfect, will be far superior to what it now has, at a cost within reach of an anaemic budget. The installation expense of the hot water cooling plant will be but a fraction of the fan and duct system. The central electrical or steam jet refrigeration unit would be considerably smaller for the former than the latter.

Many hospitals, in spite of the savings indicated in the new heating plant, will not be able to afford even the inexpensive type of air conditioning outlined. In these some form of portable equipment will be provided which can be wheeled into the room of the patient needing it as is now done with the portable x-ray and ultra violet lamp. Operating, delivery, constant temperature rooms and nurseries will have individual conditioning units, as well as a few rooms where patients suffering from conditions peculiarly susceptible to improvement under controlled temperature and humidity may be housed.

Foreign Practice

They are doing some things in Europe which, in principle, seem ideally adapted to hospital heating. These are being studied here but have not as yet been reduced to practical levels. As it is the aim to confine these speculations on the Hospital of Tomorrow to methods and principles which are now

technically and economically available or may readily be made so, these prospective developments are merely noted, rather than incorporated in the specifications.

The Sulzer system developed in Switzerland and used extensively on the Continent and to some extent in England heats water by electricity through the night, utilizing cheap hydro-electric power or current supplied at production cost by Public Utilities, during off-peak hours. The hot water is stored in insulated containers of sufficient capacity to heat the building and supply domestic hot water for 24 hours, with ample reserve.

The purpose of a heating plant is to prevent the occupants of a building from losing body heat too rapidly for comfort. The three factors for health and comfort are air temperature, humidity and surface temperature. In England particularly the third factor is being emphasized by the use of wall panels and pipes instead of radiators. These warm the walls and ceilings themselves and reduce the emission of body heat which always radiates directly to cool surfaces. In the author's experience panel heat proved far more comfortable at 65 deg than our radiator heated rooms at 70 deg. The laboratories experimenting with electric panel heating have demonstrated that if the human body does not have to give up its heat to cold surroundings men can work in comfort in the lightest summer clothing in a room where the temperature is only 55 deg. The trend of this research and the imminence of its success is indicated by what L. W. Schad, research engineer, said last June, "At the present rate for electrical power direct heating by electricity is not economical. However, I look forward confidently to an economical system using pipes in the wall and ceiling which are heated in Winter and cooled in Summer by reversed and direct refrigeration respectively."

A few engineering details of some importance may be mentioned.

Ventilating

A limited amount of exhaust ventilation will be installed in corridors for general circulation and to remove excess heat and odors from the kitchen, laundry and various plumbing units. Portable electric air purifiers, not to be confused with ozonators, will be provided where needed. These devices cleanse and deodorize the air and strengthen its ion content. They are valuable adjuncts to air conditioning, particularly in the rooms of patients suffering from maladies which produce so much odor that no ordinary ventilation is adequate.

Piping

All piping used in the building will be of non-corrosive metal, with joints welded or sweated without the use of threaded fittings. This will largely eliminate water flow noises at elbows and joints as well as the danger of leaks. With no leaks to be feared branch control valves for both heating and plumbing pipes on the various floors may be eliminated, only riser valves being used, with a further saving by the omission of pipe shafts and access doors. Horizontal mains and connections will be run under the floor slabs, easily reached through removable panels of acoustical material which form the hung ceilings. Both hot and cold water pipes will be insulated with aluminum foil air cell covering.

Plumbing

Plumbing fixtures, particularly valves, faucets and fittings, will be of the best quality and design, their longer life and lower maintenance representing real economy, compared with the competitive articles so frequently installed in hospitals. For most purposes fixtures of standard type made by the thousand will replace the special hospital basins, bowls, sinks and controls, made a few at a time at a far greater cost. There will be fewer toilets and baths which the patients use but seldom and more conveniently placed automatic bedpan washers and other time- and labor-saving devices which the nurses use constantly. All plumbing will be made more quiet.

Electrical Work

The frequent troubles with electrical conduits, from abrasion and condensation of moisture around circuits and feeders, will be remedied by the omission of all conduits. *Park Cables* will be used for the feeders and circuits in the floors and walls. These require no protection other than the sheathing which is a part of the conductors. The cables are of the sort successfully adopted in underground work, with the difference that the feeders and circuits used in the buildings will be of the single conductor type with only one wire to a fixture. This wire is insulated, its protecting sheathing acting as the return wire for the grounded side. All electrical fuses will have been eliminated from light distributing switchboard and power panels. Combination thermal cutouts and noiseless toggle switches will be used on all small wiring, while the large feeders will be protected with the usual circuit breakers. The building will be lighted with ultra violet ray incandescent bulbs similar in appearance to the old electric light bulbs.

Sterilizing

Aseptic sterilization is perhaps the hospital's most technical and critical procedure. It consists of the application of moist heat at 212 deg or more for a given period of time at a predetermined pressure, to destroy pathogenic bacteria and other contamination, in water, or on instruments, utensils, surgical dressings, rubber gloves, gowns, etc. Proper sterilizing means safety, faulty sterilization infection and perhaps death to the patient. The equipment and processes of sterilization are complex and the danger from mistakes, human and mechanical, causes constant concern to surgeons, nurses and management.

Sterilizing apparatus is made by firms specializing in hospital equipment on whose standards and recommendations as to sizes, types, methods of installation and operation the hospital and Consulting Engineer rely. Steam, electricity, gas and even kerosene are variously used. High pressure steam has generally been accepted as the most economical medium, a conclusion which probably would not be confirmed if all of the cost factors of installation, maintenance and operation could be arrived at accurately. Recently electrical equipment has been greatly improved and more generally used. But conforming to the essential principles of the steam apparatus it is overcomplicated, with many expensive and temperamental gadgets designed to control temperature and pressure, to protect the shell and heat units from damage due to burn-outs, and to record the time during which the articles being sterilized have been subjected to what temperature and what pressure.

Since our 100 bed hospital is to use electricity a comparison with steam is pertinent to the thesis:

There will be 25 sterilizing units, the factory cost of which will approximate \$6,000 for steam and \$8,000 for electricity. The installation of steam, providing 15 hp capacity in the boiler with an equal reserve, many hundred feet of steam mains and returns to sterilizers scattered all over the building, reducing valves, control valves, traps, gauges, etc., would be many times the cost of power wiring. The maintenance of the steam supply represents a constant expense, the electrical practically none at all.

The author is indebted to the Wilmot Castle Co. for the data in the comparison of electric and steam consumption for one complete sterilization in each piece of necessary equipment. While these figures are approximate, most of them are based on actual tests and are sufficiently accurate for the purpose.

Thus if the hospital had a low rate of 1c per kw for power, a not unreasonable assumption if much of the heavy sterilization of water and dressings can be done at night, at the off-peak rate, the total cost for one operation each of all units would be \$1.49. Few hospitals have accurate figures on steam production. A 100 bed hospital might generate it at 80c per 1,000 lb, at which one sterilization would cost 39c or an apparent saving of \$1.10 over electricity. However, the cost of keeping up 15 hp of high pressure steam throughout the 24 hours for an actual use of but five, plus the constant loss of steam by radiation, would undoubtedly wipe out this apparent saving several times over.

The hospital of tomorrow should be equipped with simpler and more economical sterilizing apparatus. The application of thermo-electric principles successfully used in other fields will provide simple and positive controls of steam pressure, temperature, and time; eliminate many of the present uncertainties and complicated operating procedures, and insure uniformly positive sterilization with far less delay than is now possible.

With the various changes outlined, the management, maintenance and operation of the mechanical and electrical plant will be so simplified that many of the old crew of engineers, electricians, and repair men may be removed from the payroll. The equipment will be inspected and serviced monthly by a Hospital Management Bureau at a small annual charge.

While the limitations of time and patience preclude any further prognostications, certain principles of approach may be suggested. The Building Committee, which in the past has accepted almost without question what the engineer has given it, will want to know more in advance. Questions will be asked,—is it needed, is it the best thing for the purpose, will it prove economical? In the investment field the old principle of *caveat emptor* has been superseded by the new dictum, *caveat vendor*. The burden of proof will rest with the consulting engineer, that his plant will accomplish what the hospital requires, will operate not at average, but maximum economy, be simple to keep in running order, and give every promise of long life. The consulting engineer will be a constant in the planning council and present with the architect and consultant at meetings of the Building Committee. Here he may sell quality and demonstrate the wisdom of paying 10 to 15 per cent more for the best, where the best is cheapest. Here he may render sound and con-

structive service by advising on the practical and economical approach to the mechanical innovations for which someone invariably calls in each hospital project.

The engineer will be expected to prepare a detailed forecast of mechanical operating cost for the hospital budget, and,—mark this,—under a retainer to supervise the operation of his plant for five years,—or even life,—making quarterly or semi-annual inspections and reports. How many times a fee of $250 or $500 could the consulting engineer save the hospital if he had the opportunity and authority to protect his plant from abuse and see that it was properly run!

In the hospital of tomorrow the consulting engineer will be an economist as well as a designer and by the application of sound and progressive principles will endow many beds.

DISCUSSION

James Govan [2]: Dr. Mills has shown that the mean temperature enjoyed both winter and summer over the largest portion of the United States and over a large portion of Canada is very nearly right for human comfort and well-being. One of your jobs as engineers in the near future is going to be to persuade the architects with whom you are associated to provide you with buildings that will hold themselves at approximately that mean temperature with the least amount of artificial assistance in order to simplify the problem of providing air conditioning and other plant improvements.

It is quite easy to construct buildings under climatic conditions on this continent that will hold themselves very nearly at outdoor mean temperature levels over the entire winter and also during summer months. This would then provide sufficient finances to make available mechanical equipment to produce the artificial stimulating conditions or depressing conditions outlined by Dr. Mills.

I say that advisedly because we are actually running buildings at the present moment where our heating plant has been cut 70 per cent from what would be required in a normal building.

Mr. Neergaard mentioned that it would be fine to have hot water running at about 140 deg to 220 deg. We designed a hospital in Canada that was opened in July this year, and the latest report I have on it is that in the forced hot water job, at 18 below zero outdoors, the temperature of the flow water from the heater was 128 F and it was two days after the cold spell was over before they had to raise the temperature of the water.

One of your members, H. H. Angus, has information on a hospital (in northern Ontario), for which he was engineer, where at 40 below zero the flow temperature of the water was 140 deg.

The variation in heat load in four hospitals is as follows: one burns a ton of coal a year for 550 cu ft of inside heated space; another a ton of coal for 1,500 cu ft; a third uses a ton of coal for 2,500 cu ft and a fourth a ton of coal for 3,200 cu ft. That is a variation from 550 to 3,200 cu ft per ton.

All the theoretical formulae for this type of building indicate that no such reduction in coal consumption is possible, and for that reason I say, as I said at the meeting in Cincinnati a year ago, that some of the formulae now being used for calculating coal consumption are nearly as inaccurate as the formulae for calculating

[2] Govan, Ferguson and Lindsay, Architects.

the amount of heating plant necessary when the heat capacity of a structure is almost entirely ignored. Some of the factors that enter the picture are totally ignored.

With Mr. Neergaard I agree, that in the hospital of the future it will be necessary to have some simple type of apparatus that will provide different air conditions in different rooms, because doctors, I believe, are going to ask us to provide different conditions in almost every room in the building.

How are you going to do that with a central plant? I think that the development will be along the lines of individual units in rooms, not necessarily installed in all the rooms but the result might be obtained by using portable units which would be brought into the rooms either to raise or lower the temperature, to raise or lower the humidity, or to change the ionic content of the air as the case may be, subject to the medical prescription of those who are supervising the case.

A. A. ADLER: In my judgment Mr. Neergaard's paper somewhat overstresses the economic aspects of hospital design and correspondingly underestimates the type of service which a plant is required to render. The desire for savings might be driven to a ridiculous extreme. It is much cheaper to freeze than to keep warm; ventilation might be dispensed with irrespective of the discomfort involved; lighting might be of such low intensity as to be negligible or the lighting fixtures might even be dispensed with. We do not need beds, clothes, chairs, tables, etc., unless we set a lower standard of living below which we do not intend to sag. Accordingly we are confronted with the real problem at issue: What is the minimum type of service we are expected to render? After this is known the engineering problem is to render the service at the minimum cost with due regard to reliability of service.

In institutional work there are several intangibles which are not found in industrial applications. It is easier to secure funds for the better institutions than for the poorer ones. It is human nature to subsidize the best and to abolish that considered unfit. At present most hospitals, like other institutions depending upon voluntary contributions, are feeling the pinch of poverty. Perhaps this stringency is only temporary. The remedy lies in more liberal support in good times rather than by investment of potential donors in securities when such securities reach unprecedented values.

The bulk of the services which a hospital renders is initiated by the physicians making up the executive staff. They are proud of their undertaking, and justly so. It is reasonable to suppose that they would prefer to sponsor an institution which is an advance over existing institutions. Perhaps we overdid things in hospitals as we overdid things in other fields. These are the penalties we must pay for progress. The best way to determine the limit in any enterprise is to exceed prudent limits until the majority believe that further extensions along the line are folly. We still have many with us who are enchanted more by well expressed opinions than by brutal facts.

Relative to air conditioning, savings can be made by recirculation when such recirculation could in nowise contribute to contagion. A physician who once views a bacteria under a microscope is hard to convert to a believer in recirculation. However, there is more to it than this simple prejudice (if it is that). Suppose, in an extreme case, a patient were to develop a contagious disease, while in a non-contagious ward, and by some chance it should be a malady similar to that with which the air conditioning system connects. In case of a damage suit at law it would be difficult to prove that no negligence was implied or contributed by the authorities in charge. No recirculation would remove even the slightest trace of suspicion.

In conclusion, I believe that economies in services should be given liberal but not extravagant consideration.

C. H. Turland: I agree with Mr. Adler in some of his remarks. In our Winnipeg General Hospital, built about 25 or 30 years ago, there was a very large air conditioning plant installed, and I think the total investment was approximately $35,000 or $40,000. It ran for two weeks and has never run since. It was designed, I believe, in the eastern states, laid out for conditions up there where we have to figure on 40 deg below zero outside temperature. No provision was made for recirculation. The consequence was the cost of operation was so high that the hospital could not afford to operate it.

There also seems to be a lack of cooperation with the architects in obtaining a properly insulated building. I think we should, as a Society, impress upon them the need for that. If some effort could be made in getting architects to give some consideration to the insulation value of buildings, it would greatly simplify the work of air conditioning engineers.

H. H. Angus: I think that one of the main points brought out by Mr. Neergaard is the supervision of jobs after they are installed. We have been putting equipment in hospitals, and after the equipment is installed the Board hires an operating engineer who runs everything for a few days. At the end of that time he concludes, perhaps, that part of the apparatus hasn't any particular value and decides to stop operating it. Later on something breaks or gets out of order and as he is probably operating with a minimum staff, the simplest thing to do is to shut it down. As a result it is never repaired and at the end of a year or two the plant is 15 or 20 per cent below the efficiency it is designed for and some of the ventilation equipment is lying idle.

The designing engineer could save a great deal if allowed to help operate the institution or to give advice after the installation is completed. As a matter of fact, however, I don't know of any job where that has been done in a thorough manner.

There is another case in connection with the power plant where I would suggest low pressure boilers. In smaller hospitals of say 100 beds, which take 130 or 140 hp, it is probably simplest to heat the building by hot water pumped directly through the boilers and to have a small auxiliary high-pressure boiler to generate the high-pressure steam. That makes a simple job and in many cases it is not necessary to operate the steam boiler all the time. It may only operate one or two days a week, sometimes less than that.

On the other hand, in hospitals of 500 or 600 beds, where there is a demand for a large amount of steam, the high-pressure boiler costs very little more to operate than the low-pressure one. With a good high-pressure plant you get probably 75 or 78 per cent efficiency and you cannot compete with this by purchasing electricity at one cent per kilowatthour and using it for sterilizing after allowing for standby losses, etc.

Under normal circumstances the capital cost of electric sterilizers installed, including wiring to them, will be more than for steam sterilizers, including piping, except for isolated units. There is no justification for using low pressure boilers in a large hospital. On the other hand, their use is justified in a small institution where the cost of labor would be a large factor.

Mr. Neergaard: The problem mentioned by Mr. Adler is one which I have found very general in the past 15 years: what does the hospital really want, what does a particular group of doctors want? Their suggestions are usually based on something which they have seen in operation elsewhere, with little knowledge of how it actually works in practice. My reason for recommending that the engineer sit in the planning council is an obvious one. Only on one project with which I have been associated has the engineer met regularly with the architects and Building

Committee during the discussion of the plans, and the resulting building was highly successful.

The engineer knows the importance and value of adequate thermal insulation. He knows what the various special mechanical requirements mean in terms of equipment and cost and if present can make suggestions which crystallize the ideas of the doctors and medical superintendent along practical lines.

In emphasizing economy in construction and design it is not my idea at all to build cheap, unattractive hospitals. The trouble in the past has been that too frequently the monumental buildings have been more beautiful than practical and in the light of experience, excessively costly to operate. We can have very much simpler hospital mechanical plants and save a great deal of money in the cost both of installation and operation. Remember this, that every dollar spent unnecessarily in the operation of the hospital power plant means that much less which can be spent for the care of the patient. The real job of the hospital is to take care of the patients.

No. 988

STUDY OF FUEL BURNING RATES AND POWER REQUIREMENTS OF OIL BURNERS IN RELATION TO EXCESS AIR

By L. E. SEELEY * AND E. J. TAVANLAR † (*MEMBERS*), NEW HAVEN, CONN.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the *American Oil Burner Association* and conducted at Yale University

PREVIOUS studies on oil burners and heating boilers have revealed performance characteristics such as: boiler efficiencies at different outputs; the effects of furnace drafts, intermittent operation, excess air on output and efficiency, comparisons of oil and gas, measurements of sound, etc. Much of the above was necessarily determined according to certain fixed operating standards in order that the comparisons might be entirely valid. None of the work has adequately revealed, however, the complete operating range of an oil burner as it relates to maximum and minimum oil-burning rates with varying proportions of excess air.

Previous standard tests, called Series A tests, have always revealed well-defined maximum and minimum fuel-burning rates. Testing standards called invariably for clean combustion at 10 per cent CO_2 (or 50 per cent excess air) and a furnace draft of 0.02 in. of water. It seemed evident that, depending upon the operating conditions selected, the maximum and minimum fuel-burning rates would continue to be equally well defined though probably of a different order. It was thought worth while to investigate this phase of burner operation completely since it would provide a more comprehensive picture relative to oil burner capacities and it might also reveal certain limiting factors which might or might not be inherent in the burner itself.

It was also felt that such an investigation would raise some questions in regard to capacity listings of oil burners without qualification as to operating conditions and it should also suggest desirable standards in the event of adopting qualifying standards of operation.

The upper limit in fuel-burning rate might be determined by: (1) oil supply; (2) air supply; (3) furnace size and/or furnace design or; (4) undesirable

* Assistant Professor of Mechanical Engineering, School of Engineering, Yale University.
† Graduate student, Graduate School, Yale University.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934, by L. E. Seeley.

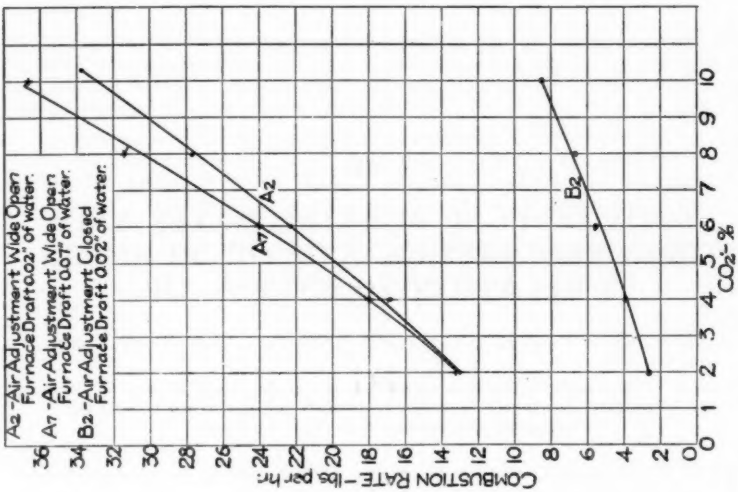


FIG. 2. FUEL-BURNING RATES v. PER CENT CO_2

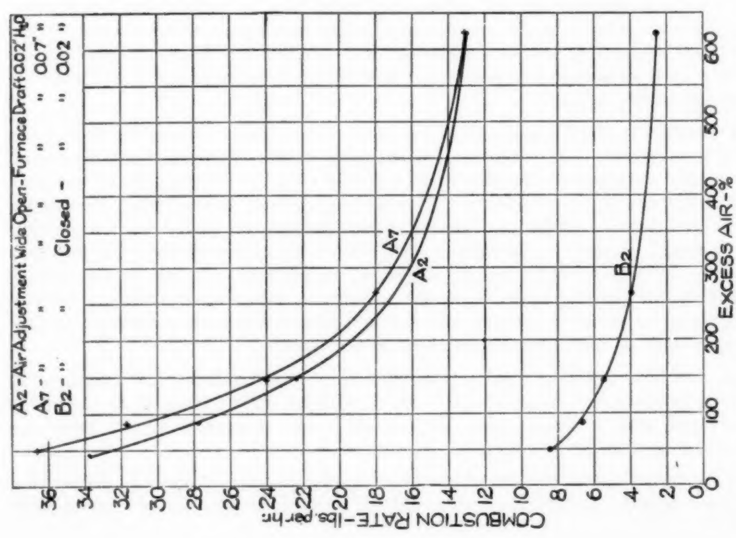


FIG. 1. FUEL-BURNING RATES v. PER CENT EXCESS AIR

combustion vibrations or sounds. These factors inhere in the device itself or in its installation. Additional limiting factors might be due to the boiler, such as: (5) small furnace or firepot volume; (6) excessive draft loss due to small flue passages and/or long flue gas travel. These factors are extraneous to the burner itself as would also be the limitation induced by a chimney of insufficient capacity. These latter factors are too uncertain and variable to consider at this time although their possibilities should be recognized.

The lower limit in fuel-burning rate may be determined by such factors as: (1) degree of fuel atomization; (2) effectiveness of mixing oil and air, and (3) furnace size and/or design. It is assumed that conditions necessary for satisfactory ignition are present. While an evaluation of these factors is somewhat more speculative, it is believed that actual tests may show some interesting characteristics according to types of oil burners.

An example of capacity tests is shown by Fig. 1. Curve A_2 shows the fuel-burning rates possible with the maximum supply of air available at various percentages of excess air. The air supply in this case was provided entirely by a fan; the air adjustment was kept wide open and the furnace draft was maintained at 0.02 in. of water. The decrease in fuel-burning rate is very marked between 50 and 250 per cent of excess air and less marked above 250 per cent.

Curve B_2 represents the least quantity of oil that could be burned at various proportions of excess air. In this case the air supply was reduced to its minimum value by means of an air adjustment provided on the burner. This leaves some uncertainty as to whether or not the fuel rate could have been further reduced by a more positive air adjusting device. A furnace draft of 0.02 in. of water was maintained in these tests.

Curve A_7 was similar to A_2 except that the furnace draft was raised to 0.07 in. of water. It should be pointed out in connection with comparisons of Curves A_2 and A_7 that every flue gas analysis has some degree of error due to air leakage into the boiler. While air leakage is kept as low as possible, because of this fact, it is necessary, for really comparative tests, to keep this air leakage constant. This is done by maintaining the same furnace draft for each series of tests. A comparison of two series of tests at different furnace drafts involves some unknown change in the air leakage rate. The exact curves therefore would probably be closer together than those shown here.

The range in fuel burning rates at 50 per cent of excess air appears to run from 8.4 to 32.8 per hour. The greatest overall range exists at the lowest percentage of excess air shown (*i.e.,* 50 per cent excess air). At 625 per cent of excess air the range is from 2.6 to 13 lb per hour, a spread of 10.4 lb compared to 24.4 lb at 50 per cent excess air.

Horizontal lines drawn through 8.4 and 13 lb, respectively, would enclose a zone where the rates of fuel burning between the above values could be achieved with any proportion of excess air from 50 to 625 per cent. On the other hand, the possible variation in excess air for a fuel-burning rate of 26.6 lb, for example, would be between 50 and 100 per cent. Burners chosen to operate somewhere near their maximum fuel-burning rates would practically dictate adjustments calling for low percentages of excess air.

While high fuel-burning rates would generally require decreases in excess

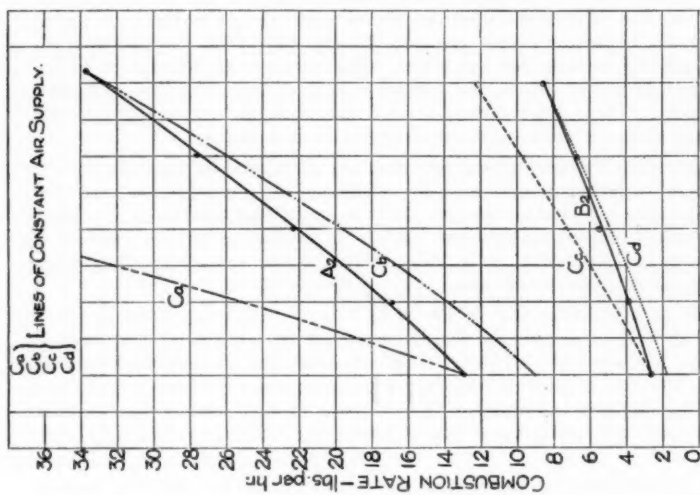


FIG. 4. FUEL-BURNING RATES v. PER CENT CO_2, AND LINES OF CONSTANT AIR SUPPLY

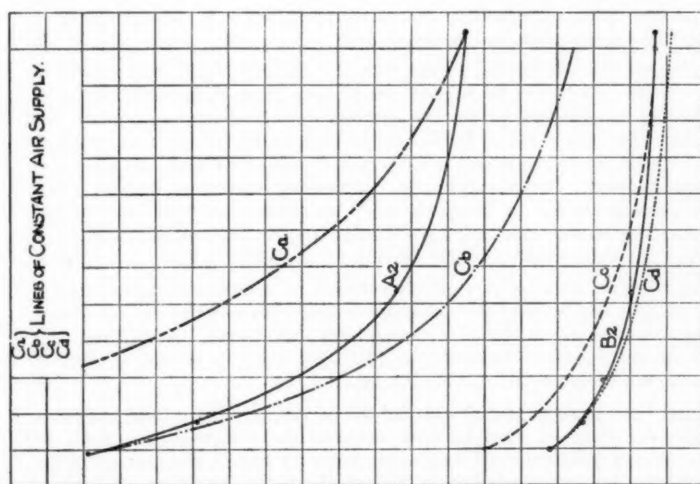


FIG. 3. FUEL-BURNING RATES v. PER CENT EXCESS AIR, AND LINES OF CONSTANT AIR SUPPLY

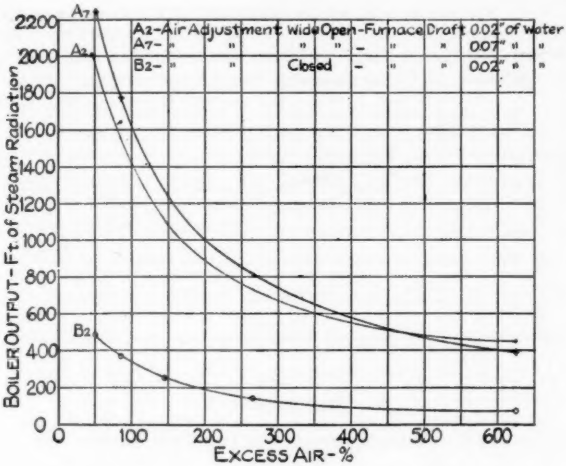


FIG. 5. BOILER OUTPUT v. PER CENT EXCESS AIR

air, low fuel-burning rates actually require increases in excess air. The penalties in the way of low efficiency at low fuel-burning rates are strikingly evident. The extreme range of this oil burner runs from 2.6 on Curve B_2 to 33.7 lb on Curve A_2, curve A_7 not being considered because of its different furnace draft. It is obvious that a range from 8.4 to 33.7 lb would be economically more

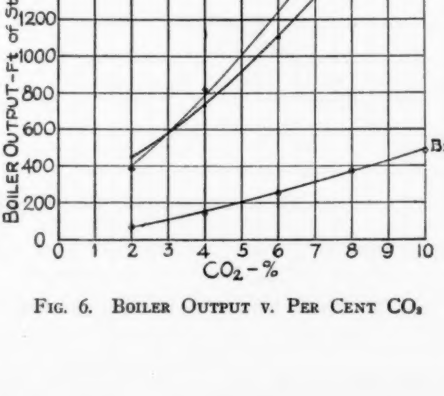


FIG. 6. BOILER OUTPUT v. PER CENT CO_2

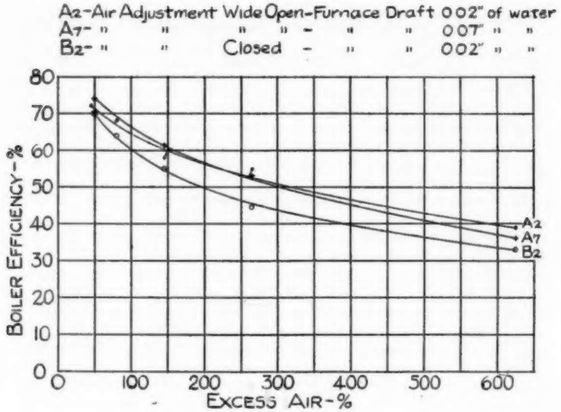


FIG. 7. BOILER EFFICIENCY v. PER CENT EXCESS AIR

desirable because any fuel-burning rate between these values could be managed with only 50 per cent of excess air.

Fig. 2 shows the results plotted with CO_2 instead of excess air. It will be noted that 2 per cent CO_2 is obtained with an excess air of 625 per cent and 10 per cent CO_2 at 50 per cent excess air. The range from 2 to 10 per cent CO_2 seems ample although probable values above 10 per cent CO_2 may be estimated by extending the curves. The extension of the curve beyond actual test points is not recommended, however.

Certain interesting things may be shown in other cases that do not appear in these tests. It might happen with another burner that the fuel-burning rate

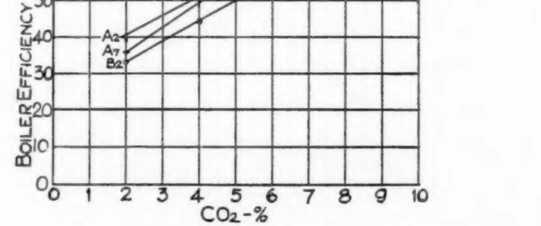


FIG. 8. BOILER EFFICIENCY v. PER CENT CO_2

with maximum air could not be carried all the way to 10 per cent CO_2. This might mean a limitation possibly due to having reached maximum fuel capacity or due to operating difficulties caused by furnace proportion or design. The factor or factors contributing to this limitation could be found by study and a better balance of oil, air and furnace design achieved.

Again, limitations in the low fuel-burning rates might be traced to fuel preparation or to mixing or to furnace design. Furthermore, if standards of operation are ever adopted, curves like A_2 and B_2 would establish the boundaries for determining the operating range of any particular burner.

Figs. 3 and 4 show a very interesting characteristic which may or may not be common to other mechanical draft burners. Curves A_2 and B_2 in Figs. 1

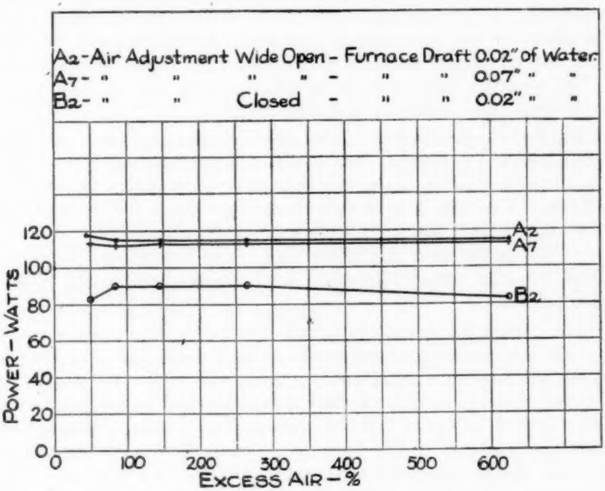


FIG. 9. POWER REQUIREMENTS v. PER CENT EXCESS AIR

and 2 were obtained by wide-open and completely closed air adjustments, respectively. The supposition would ordinarily be to the effect that the air supplied by the fan would be practically constant for each air adjustment. Curves C_a, C_b, C_c and C_d are lines of constant air supply.

Curve C_a was based on the air supply actually available at the point of maximum excess air on Curve A_2. C_b was based on the air supply actually available at 50 per cent excess air on Curve A_2. C_c and C_d were similarly determined for corresponding points on B_2. These curves merely indicate the fuel-burning rates that should be attained at various percentages of excess air if the air supply (*i.e.*, pounds per hour) is the same in each case. The curves are merely based upon combustion calculations.

These clearly show that, with an increase in excess air and the resultant decrease in fuel-burning rate, the air supply for Curves A_2 and B_2 actually do not remain constant even though the air adjustment is untouched.

A possible explanation for this would be that the density or volume of the fuel vapor might act as a barrier or in some other manner to reduce the flow areas ordinarily used by the air. That some throttling or holding back of the air takes place seems evident enough. The result shown by Curves A_2 and C_b indicates that if the air supply necessary for a combustion rate of 32.8 lb per hour at 50 per cent excess air had actually remained constant, a combustion rate of only 7.1 lb per hour would have been possible at 600 per cent excess air instead of the 13.1 lb actually obtained.

This suggests that if fan capacities are selected from regular fan test data, the volume of air available in actual operation might be seriously reduced and the anticipation as to probable maximum capacities thwarted. Curve C_a indicates what might have been expected from calculations as to maximum fuel-burning rates and Curve A_2 shows what would actually happen. Whether or not the effect of the fuel vapor, its method of delivery, and its combustion, is to reduce the air supply in every case as much as shown in these tests is not known. The question does arise as to whether tests of the character reported are valuable both experimentally and commercially.

Figs. 5 and 6 show the curves A_2, A_7 and B_2 plotted in terms of boiler output in feet of radiation (1 ft-rad = 240 Btu per hour) against excess air and per cent CO_2, respectively. This merely indicates what these fuel-burning rates mean in terms of actual heat output from the boiler. It should be clearly understood that boiler output values will depend upon the heat absorbing efficiency of the boiler and that these same tests conducted in another boiler might show different values and even some modification in the trends of the curves themselves.

Figs. 7 and 8 show the actual efficiencies obtained in these tests plotted against excess air and per cent CO_2, respectively. Fig. 9 shows the power requirements in curves for the fuel-burning tests reported in this paper. The curves, A_2, A_7 and B_2, are the power curves for the corresponding fuel-burning curves similarly designated. It will be noted that A_7 requires less power, probably due to the lower discharge pressure in the furnace. Curve B_2 shows that less power is required where the air supply rate is reduced. The fact that all of the curves are practically horizontal may or may not be typical. It is true that in following the curves from left to right the fuel rate decreases and the air rate increases and thus one factor may compensate for the other and the power remain practically constant. Actually, the power requirements of various oil burners are so very different, even at similar fuel-burning rates that generalizations are especially inconclusive.

CONCLUSIONS

1. Maximum fuel-burning range of an oil burner is obtainable, according to fixed operating standards, when the standard requires a low excess of air (or high CO_2).

2. The maximum fuel-burning rate at low excess air may be increased by increasing the furnace draft.

3. The possible variation in operating standards (*i.e.*, per cent of excess air) is reduced at the high fuel-burning rates. In other words, more economical fuel-burning conditions are assured.

4. There is a certain range in fuel-burning where it would be possible to have any excess of air from 50 to 625 per cent. The probable economy of fuel burning in this range would be highly speculative, assuming the absence of any standard of operation.

5. Very low fuel-burning rates make increases in excess air mandatory. Efficient fuel burning becomes virtually impossible.

6. The air delivery from the fan of the burner used in these tests falls off with an increase in the fuel-burning rate, even though the air adjustment remains fixed. To what extent this is a characteristic of all oil burners is not definitely known.

7. Any statement in regard to fuel-burning rates of an oil burner should preferably be qualified by operating standards relating to excess air (or CO_2) and furnace draft.

DISCUSSION

D. W. NELSON (WRITTEN): The testing program on oil burners, the report of which is being closed in this present paper, points out the important fact that the high efficiency of combustion as shown by 10 per cent CO_2 is a reasonable standard to hope for. This has not been generally known among oil burner dealers. Two years ago the placing of a requirement of 10 per cent CO_2 in the specification for a large oil burner installation prevented several dealers from bidding. They considered such an efficiency of combustion almost impossible of attainment. It seems that most installations are made without the benefit of CO_2 analysis and the efficiency of combustion is left largely to chance. A high fuel cost is usually explained by the boiler or furnace being inefficient, which is true in many cases, but in others the fault is in the burner installation or adjustment.

A survey is being made of oil burner installations in Madison, Wis., to determine costs and efficiency of operation. At the present time approximately 40 installations have been investigated. The average CO_2 determined in the combustion space was 8.1 per cent and at the smoke pipe connection 7 per cent. The average flue gas temperature was 524 F.

Table A shows these results.

TABLE A

CO_2 Per Cent	In Combustion Space	At Smoke Pipe Connection	Flue Gas Temperature	No.
Below 5	..	2	Above 700	10
5–6	..	8	600–700	6
6–7	6	8	500–600	6
7–8	5	6	400–500	9
8–9	7	7	300–400	2
9–10	4	3	200–300	4
10–11	5	3	Below 200	3
11–12	..	1	....	..
Total..........	27	38	....	40

These figures show that in actual installations few burners operate at 10 per cent which has been found in laboratory tests to be of practical attainment. In some cases the efficiency can be improved by simply adjusting the air but in many cases the installation is not suited to 10 per cent CO_2 attainment without smoking. In one particular case the combustion chamber was rebuilt three times in three years by three different dealers in an attempt to lower excessive fuel costs. At no time was

a flue gas analysis made and the only check on the installation was the yearly fuel costs. With respect to one common type of burner some manufacturers favor a large chamber, others a much smaller chamber. Some favor a high target wall corbelled out to secure better mixing; others a lower and straight wall so as not to mix products of combustion with fresh air and fuel. Since these various burners are almost identical, it would seem that one certain design should be better adapted. In some installations made according to manufacturer's rules, it seems impossible to secure over 60 per cent without smoke. Too often the installer clears up the fire by a large amount of excess air and blames an excessive fuel cost on poor heat absorption.

The flue gas temperatures recorded show evidence of probable lack of heat absorption. Ten out of 40 installations tested showed a temperature above 700 F. Twenty-two out of 40 were above 500 F. Five of the seven with a temperature below 300 F had additional heating surface in the form of heat savers. However, the high flue gas temperature is not all chargeable against lack of heating surface. In many cases the rate of firing is too high. High heat release together with large excess of air as shown by low CO_2 causes high stack temperatures. By using a lower oil burning rate and less excess air, the temperature at the exit can be greatly reduced. In one particular case a draft control was installed to reduce fuel costs, yet at the same time the oil burning rate was left too high, resulting in an unnecessary 760 F flue gas temperature. In another case the placing of a simple baffle to direct the gases over the direct heating surface lowered the temperature 100 F.

In a seven room residence not insulated, where careful daily records have been kept during this heating season, close to the smallest burning rate for the burner is used. The rate is about 1½ gal per hour. On the coldest day this year the burner ran 8.9 hours. On an average day it runs about 3 hours. The air adjustment is choked down to suit the low fuel rate yet the highest CO_2 obtainable thus far is 8 per cent. The burning rate is too high for the heat losses of the house, which results in a flue gas temperature of 500 to 550 F. It would be desirable to further reduce the fuel rate, yet the present paper indicates that greater difficulty would be met in obtaining reasonable efficiency. The answer seems to be that burners are too large for average small residences and that a gain in efficiency would result from the careful design and installation of small units. The problem of proper fuel preparation, mixing and furnace design are intensified in the smaller sizes. The manufacturer of this burner reports little difficulty in getting a satisfactory fire in the larger size burners, but for unaccountable reasons often obtains poor results in small installations. It is unfortunate that in an average residence the heat losses are such that a burner cannot be installed near its maximum rate of burning, where the present paper shows there is little chance for high excess air and its attendant low efficiency.

The present paper as well as the others of this series should result in improvement of burners and installations. Installers should be educated to the significance of CO_2 analysis and flue gas temperatures. Burners should be rated and sold with a guarantee of a certain CO_2, such as 10 per cent, and this at a certain fuel burning rate.

W. K. WALKER: For the last few years I have been doing some research work and there is one point which I can't help noticing has not been brought out. When steam is generated at the boiler outlet and discharged into the mains in response to a demand from the thermostat, the rate of discharge seems to have a very great effect on the annual fuel consumption.

In other words, with a low rate of discharge of steam, consider an extreme case in which the oil burner might run for hours and hours attempting to satisfy a thermostat which is located at the far end of the building, as in the case of an installation where the steam doesn't reach the radiators at the far end of the building and yet is overheating the rest of the building.

We have found that in such a case there is an added fuel consumption of 30, 40 or 50 per cent. The difference is great. You can increase the flame and get better results. But if you increase the flame or rate of flow to a point where, when the flame comes on or the steam is turned on, you quickly get the steam right down to the end of the line, then you get even heating through the building, you satisfy the conditions, and you shut it off. That is a very important point and I believe that in studying fuel consumption on buildings, we should compare CO_2, etc., only where the CO_2 affects the amount of steam which is generated and discharged from the outlet of the boiler per unit of fuel, that is, as it affects the boiler efficiency, but the amount of steam and the rate of discharge and the size of the flame is another question and that does affect the annual fuel bill. So I think there are two questions, CO_2 affecting boiler efficiency, and the rate of discharge affecting the efficiency of distribution.

W. H. Carrier: I am not in the oil-burner business, but it always appeared to me that to design a burner by one organization and a boiler by another organization independently and without relation to each other was a fundamentally wrong engineering procedure. I don't know whether I am stepping on anybody's toes or not, but eventually the thing that is right from an economic and engineering standpoint is going to prevail, whether any particular group of manufacturers like it or not. That means air conditioning and the boiler business and the burner business or any other. True economic consideration will in the end prevail. We might as well face it openly and recognize it. I think today that the manufacturers of boilers are seeing that and the leading manufacturers are building a unit which is engineered with respect to burner, combustion chamber and surface in such a way that they are able to give the maximum results and dependable results, which I believe will do much to advance the oil-burning industry.

When you buy an automobile you don't buy the engine from one person, the chassis from another and the body from some one else, and put them together. It is bought as an assembled unit. In the small boiler field I believe that is bound to be the ultimate trend. I believe that it is possible to put in units which will give efficiencies not of 70 per cent but of at least 80 per cent and to do it dependably, regardless of any normal variations in draft condition, and free from smoke and other things that have been condemning oil burners.

The economic point at which certain refinements will be added will depend upon the size and upon the cost of such refinements as compared with fuel saving. Many times I believe it doesn't pay to go as far as the ultimate fuel economy, especially if you consider that maximum loads are not a condition that prevail, and where there are provisions for modification of the maximum load conditions according to weather. Very frequently a unit should be selected under size, but capable of overloading for extreme conditions, because they occur so few times in the year that they are not of economic importance, while the average load, or perhaps a little better than the average load, is the one to be met with maximum economy.

I believe that the trend toward oil-burning equipment has to be a unit of design, rather than trying to engineer this and that firebox to suit this and that burner. I realize that the burner people have the problem of replacement, which outweighs the new installations, and they are quite justified in exploiting a market for this large field, so that the oil-burner business, independent of the boiler business, has a real field in replacements.

On the other hand, there are many old installations where the consumer would be far better off to discard his old boiler completely and put in an entirely new outfit. In other words, spend more money, which will help the industry.

W. H. Driscoll: To carry Mr. Carrier's argument to its logical conclusion with regard to the engineering considerations involved and the necessity for combining

the various problems and solving them as a whole, why wouldn't it be necessary to design a heating system as a whole and engineer it, design it, manufacture and install it as a whole. If we ever reach that point we will all be in the employ of Mr. Morgan or Mr. Mellen.

MR. CARRIER: That is quite possible, Mr. Driscoll, providing houses are standardized so that the same heating system can be put in every house. Then engineering consideration would be eliminated. A packaged unit would be purchased in a certain size for a certain house, but I don't believe we are ever coming to the point where we are going to standardize our dwellings.

E. J. TAVANLAR: From practical experience with oil burners for a number of manufacturers who used my services in checking their installation work, I can say that the findings in Wisconsin have been largely confirmed. There are, of course, exceptions to the statement that it is impossible to attain 10 per cent CO_2 in service. It has been obtained and I think will be obtained more often in the future. Recently I worked on an installation where the burner and boiler were old and the whole boiler setting was leaky. I went on this job with an Orsat and a thermometer to measure the flue temperature. I brought with me a 600 deg thermometer and when I placed it in the stack it became broken. After bringing in another thermometer, the flue temperature was measured to be 760 deg which, as you know, is very high. The CO_2 reading was 3.6 and I calculated that the unit was operating at an overall efficiency of approximately 50 per cent.

I must state that I didn't get 10 per cent CO_2 on this job, but I redesigned the combustion chamber, plugged up the leaks in the boiler and lowered the burner, so that the lower section of the boiler was exposed to the flame. I installed a draft regulator in the stack so that the draft could be set and maintained at 0.05 in. of water. After these changes were made I ran another test on this unit and found that the CO_2 was 7.8 and the efficiency increased from 50 per cent to about 62 per cent, if the losses to the air in the cellar are considered as part of the useful heat.

Thus, on an installation where the CO_2 is very low, it is best not to set 10 per cent as a standard, but to try and improve the job. We are saying here that 10 per cent is a very good standard and we have already given evidence to show that it is so, but it is foolhardy to expect to make a poor installation perform an almost miraculous task of attaining 10 per cent CO_2.

Perhaps not enough emphasis is given to the design of combustion chambers in relation to oil burners. To my way of thinking, a combustion chamber should help in atomizing and vaporizing the oil. If a combustion chamber doesn't do that, it might just as well not be there, and the entire boiler might as well be exposed to the flame. The essential point to keep in mind in designing a combustion chamber, which will help in vaporizing the oil, is to have the oil that is sprayed into the chamber come in contact with the hot refractory at all times. If that is accomplished, I am sure that a much better combustion condition will be found in the furnace.

Another point that I have learned in my work with dealers around New Haven is to provide a source of excess air which can be regulated. This pertains largely, exclusively in fact, to what we call gun-type burners. In gun-type burners, as you know, the oil is introduced into the chamber through a gun and the air is usually introduced in what is called an air tube. Ordinarily installation men install these burners in a boiler without providing an independent source of secondary air. By an independent source of secondary air is meant a source which can be regulated. Leakage as such is not considered secondary air. A good way of introducing secondary air in a gun-type installation is to put a flooring just below the burner of brickwork, in the nature of that found in producer-gas installations. This brickwork will ordinarily be at a high temperature because it is close to the flame. A connection leading to the outside is required which can be regulated not only in size

but also in direction, so that the quantity of air entering the chamber, independent of that delivered by the fan and going through this checkerwork, can be regulated with a damper or air shutter at the front of the boiler and its direction can be changed by changing the direction of the bricks. If it is found that the tail end of the flame is smoking, relatively more air should be allowed to come in at the tail end of the combustion chamber.

The results of my work in connection with gun-type burners have shown that this method gives satisfactory results.

The statement brought out by Professor Larson regarding high temperature and low CO_2 doesn't mean very much to me, because I don't know the combustion rate at which the burners were operating. We have shown in previous papers that there is a relationship between fuel-burning rates and flue temperatures. That is to say, if you plot flue temperature versus fuel-burning rate, it would be a straight line function, so that a statement about high flue temperatures should be accompanied by the rate of fuel burning.

The same principle applies to CO_2 although not to the same extent. That is to say, when the statement is made that 10 per cent CO_2 cannot be attained with a burner, a qualifying statement should be added with reference to operating at a certain fuel rate, because, as you probably have noticed, the per cent CO_2 varies with the same setting of air, and with the quantity of oil that is being burned.

LOUIS NORTHON: About a year ago at a meeting of the New York Chapter Bassett Jones referred to the measurable truth as the only value in science. I am very glad that Professor Seeley's charts indicate what I have experienced recently and many times previously in my practice in conjunction with oil fired steam and hot-water boilers.

A few weeks ago I was called by one of my clients to a large church, where the oil consumption of two steam-boilers called for 56,000 cu ft of fresh air supply per hour. The volume of the boiler room plus the possible air infiltration was only 14,000 cu ft in this case; therefore, only 25 per cent of the necessary fresh air supply, and because of this poor air supply, the combustion of the fired boilers was very poor. Furthermore 52-55 per cent more fuel-oil had been used than was required for combustion with sufficient fresh air supply.

Day after day, several pounds of carbon were found on the bottom of the otherwise properly built combustion chamber of the two boilers indicating poor combustion.

To overcome and eliminate this troublesome and expensive operation of the two oil burners, I designed a new fresh air supply duct. One end of this duct, properly protected, terminates in the church yard and the other end near the boilers. The size of the duct is 36 in. x 24 in. and it is equipped with a manually operated damper control.

The relatively large duct is due to gravity operation of the fresh air supply duct, having an average temperature difference of 40 F between outdoor temperature and boiler room temperature, and this resulted in an average draft of 0.02 in. of water column at the fresh air outlet in the boiler room.

After completing installation of the new fresh air supply duct, the two oil-burners were put in operation, and by using the Orsat apparatus, contrary to the first analysis of the stack gases, I obtained 12 per cent of CO_2, none of CO, and none of O_2.

The stack temperature being 580-592 F, proved that the temperature in the combustion chamber did not reach the critical temperature with noiseless combustion, and with the presence of carbon monoxide. This checked the reading of the Orsat apparatus, as previously mentioned.

The supplied fresh air reached the 230-232 cu ft per pound heavy-oil per hour, and this amount is equivalent to 25-27 per cent excess air. This resulted in a fairly good combustion efficiency with an average of 68-70 per cent.

A few days later when the outdoor temperature was 8 deg higher than at the time of the previous test, the measured excess air supply was only 22.5 per cent, and the lower percentage, of course, was due to the higher outdoor temperature. Therefore, a lower velocity was found in the gravity type fresh air supply.

A practical field test in this case checked remarkably well with Professor Seeley's charts.

It is almost impossible for a designing engineer to select a type of boiler which has been so designed for oil combustion, because of financial consideration; especially, when in an existing building the previously coal fired boilers are to be converted for oil firing, as in the case mentioned. As a matter of fact, the two boilers in question are smokeless type cast-iron sectional boilers, each boiler having 448 sq ft of actual heating surface.

The results of the previously described survey indicated that with proper combustion chamber, and suitable fresh air supply it was possible to secure, even in a smokeless and coal type boiler, an efficiency which, in practice and in my estimation, is a fairly good one.

My experience on several occasions has shown that it is good practice to take into consideration the necessary amount of fresh air supply and the volume of the boiler room, including the possible cold air infiltration. By considering these three factors I have almost never found the volume of the boiler room to be in strict accordance with the necessary fresh air supply.

Financial conditions very often compel an architect to determine the size, and volume of a boiler-room of insufficient capacity, but this might be overcome easily by an engineer, who by designing a simple and relatively inexpensive fresh air supply of the gravity type, can secure proper and economic combustion.

L. E. SEELEY: I would be inclined to make a very brief reply to Mr. Walker and Mr. Carrier and Mr. Driscoll. I think they are all suggesting that the scope of the work perhaps should be increased by a study of systems or a very special study of a particular boiler and burner.

I do think if one starts an investigation on the basis of maximum and minimum rates that they will run into their limiting conditions and can find them. That is they can start at that point and modify furnace design and see if they can continue to fill out the complete curve. I think work of that sort would be possibly a starting point in a real study of furnace design.

I want to thank Professor Larson for reading the survey. The earlier surveys made in other communities seem to indicate that the standards are improving.

No. 989

DESIGN AND VALUATION OF CAST IRON DOMESTIC HEATING BOILERS

By Charles W. Brabbée * (*MEMBER*), New York, N. Y.

EIGHT years ago the writer had the opportunity and honor to present some thoughts concerning the valuation of radiators in relation to human comfort. The idea has been further developed and several Universities are now working in this field. The Pierce Foundation Laboratory for Hygiene at Yale, completed a few months ago, also will conduct investigations of this character on heaters of different kinds.

Today it is possible to give the results of some fundamental boiler studies conducted during the past 25 years both here and abroad. It has been decided to open up the records of our company and present in broad lines some thoughts about boiler design and valuation, in order that the facts may be studied and by so doing possibly further advance the art. The field is so vast that, for the time being, a boiler group which is usually classified as *cast iron domestic heating boilers for solid fuels,* has been selected for study though some remarks may refer to other products.

In some instances it has been customary to base the value of a boiler on its grate area, which is determined by grate width and length. Many years ago a European scientist, Engineer Barlach, made exhaustive studies of various combustion chamber designs which can be classified according to the three sketches in Fig. 1. He found that the design *B* with straight walls gave better combustion conditions than a wider grate *A* and that in turn the design *C* was better than *B*. Barlach went even so far as to design industrial stoves with only one grate bar, as in Fig. 2, and found that they burned wet fuels advantageously and gave better all around results. He attributed this fact to the high velocities of air and gas at the grate and to the complete spreading out of the combustion gases in the considerably widened fuel bed. Similar improvements were soon applied to European tile stoves. They often extended to the ceiling of the room and developed combustion efficiencies of 90 per cent. However, they concentrated their heat in the upper part of the room whereas the living zone was cold and uncomfortable.

It was in 1920 when the writer investigated such apparatus at the University of Charlottenburg, establishing at that time the measurement of the room temperature at three levels, namely, ceiling, eye and knee height.

In contrast with those old stoves, about 15 years ago tile stoves were designed

* Director, Institute of Thermal Research, American Radiator Co.

Presented at the 40th Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., February, 1934.

in which the gases are first forced into a revertable flue, warming the lower tiles which face the room and then pass through an upflue into the chimney. Such stoves which used pinched grates to a considerable degree had not only a high combustion efficiency but also a high *Useful Output* into the living zone.

With open fireplaces it is often possible to prevent discharge of smoke into the rooms and to obtain better heating effects by using stones, according to Fig. 3, which perform the same duties as pinched grates.

American engineers, designing boilers, apparently had a variation of such an arrangement in mind, namely, to locate the lower nipples conveniently (Figs. 4 and 5) without increase of the boiler's overall dimension B, which changes to B_1 if straight combustion chamber walls are used. However, a pinched grate

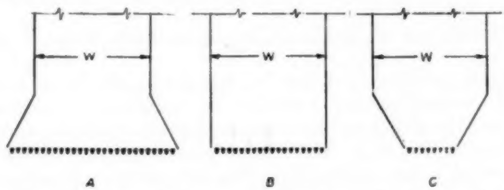


FIG. 1. TYPICAL DESIGN OF COMBUSTION CHAMBERS

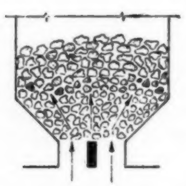


FIG. 2. BARLACH GRATE

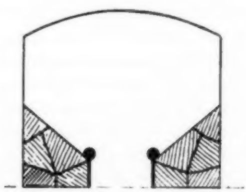


FIG. 3. IMPROVEMENT ON OPEN FIREPLACES

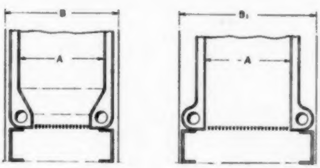
is only one item of many which influence a boiler's operation and there are several other points which should be considered, for instance: (1) total and free grate area, (2) construction of the grate bars, (3) dimensions of grate teeth, (4) the effect of teeth on front and back sections, (5) the arrangement of two grate bars in one center section, etc., but all this would distract attention from the main points.

Now consider the second important factor of grate area, namely the length A of the boiler in Fig. 6. Sometimes this dimension A is not measured directly but is taken as the lengths of the boiler L minus an arbitrary figure. Suppose the boiler of Fig. 6 is to be compared with the boiler of Fig. 7, the front and back sections of which are increased in depth so that the grate length A is the same, while the boiler length L_1 is now greater than L. Nothing has been changed in regard to actual performance fundamentals and yet boiler Fig. 7 would call for a higher rating. On the contrary, the latter design is less desirable since the water content of front and back sections and also their iron weight have increased, which are disadvantageous to the boiler's action.

Going a step further, is the grate area of a boiler actually the most important item to be considered in boiler performance? Suppose the boiler in Fig. 8 is

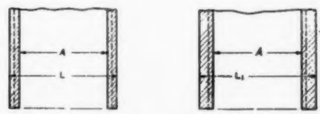
filled with coal from the grate line up to a certain height and this boiler is then connected to a given chimney. The hot gases in the chimney form a hot column *H* against a cold column *C* of the outside air and of the same height *H*. The cold and heavier outside column forces gases out through the chimney.

It is, therefore, illogical to say that the chimney is drawing gases, it is rather necessary to state that the chimney delivers the gases which the cold outside air pushes through boiler and chimney. It will be observed that a given force created by the relative action of the hot and cold column, usually called the draft, must overcome not only the resistance of the grate but also all other flow resistances, especially the one of the coal layer, which is of the utmost



FIGS. 4 AND 5. PITCHED AND STRAIGHT BOILER WALLS

FIGS. 6 AND 7. LENGTH OF BOILERS DIFFERENT BUT COMBUSTION CHAMBER EQUAL



importance. From these facts, we have justly to conclude that it would be appropriate to replace in this respect the grate area by the average area of the combustion chamber beginning at the grate line and ending at the top of fuel bed. As a more convenient and yet correct procedure the area of the combustion chamber taken at a certain height above the grate may be used.

The heat transmission in the combustion chamber is mainly by radiation from the hot firebed, whereas the influence of convection and conduction is often of lesser importance. Let us consider that the heat absorption of the firepot would be by radiation only, then flat combustion chamber walls, according to Fig. 9, and corrugated ones, shown in Fig. 10, will produce the same heat transmission as long as T_1 (high temperature) T_2 (low temperature) and the apex angle α of the radiation cone are the same.

In consequence, fins in a firepot, will be found ineffective for heat transmission by radiation. However, fins and corrugations do have a beneficial action by conduction and convection and they may improve the supply of secondary air, thereby diminishing incomplete combustion.

When using oil burners with coal boilers, the heat transmission by radiation is often considerably decreased which explains, at least in part, some of the difficulties encountered with these installations. In all kinds of boilers and especially in those with restricted radiation in the fire-chamber, the heat transmission by convection and conduction is most important and governs the design and arrangement of the flue heating surface.

If hot gases are carried in a straight pipe it will be found that the gases cool off to a greater extent on the outer surfaces and remain relatively hot in the center resulting in poor heat transmission. Where stove pipes are arranged with an L, undoubtedly by driving the stove at a high output it becomes red

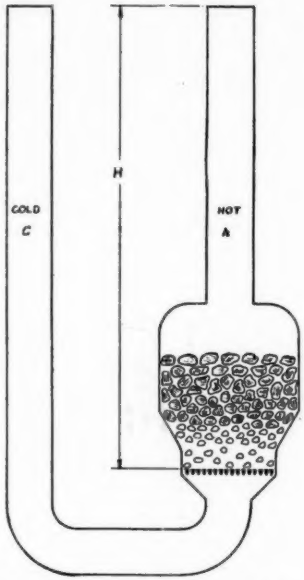
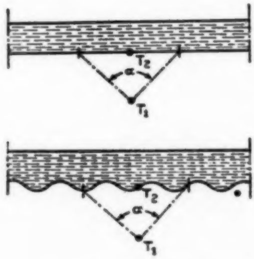
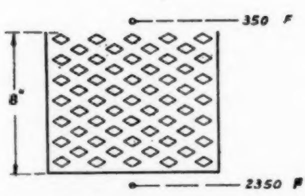


FIG. 8. DIAGRAM OF CHIMNEY ACTION





FIGS. 9 AND 10. RADIANT HEAT TRANSFER WITH STRAIGHT AND CORRUGATED WALLS OF THE COMBUSTION CHAMBER
FIG. 11. HEAT TRANSFER WITH FINS IN FLUES

TABLE 1. INFLUENCE OF DRY AND WET HEATING SURFACES

Test No.	Average Approximate Output, Sq. Ft. Steam	Average Water Line, Inches	Duration of Test Hours	Average Efficiency %	Remarks
1	1830	47	9.27	78	Pickup
2		57	9.61	78	State
3	1810	47	9.27	83	Steady
4		57	9.61	83	State

hot, indicating that very effective heat transmission takes place at that time. These facts disclose the secret of good heating surfaces. They must be so designed that in no way can stratification of hot gases take place, while the ideal must be to bring the maximum amount of hot gas particles into proper contact with the heat receiving surfaces. This rule is always important but most vital where only a comparatively small part of the total heat transmission takes place in the fire-chamber and, therefore, especially on oil boilers. Though the flues in Fig. 11 are comparatively short, they reduce the combustion temperature from about 2350 F, without any trouble, to 350 F. We have seen many visitors astonished when they look in the fire-box with its brilliance, and then touched the relatively cool smokepipe.

In fact, in some cases a part of this heating surface had to be eliminated because the gases cooled down too much and would, especially on water boilers and at low loads, come below the dew-point. This produced undesirable condensation in smokehood and chimney. Again another point has to be taken into consideration, namely, the cleaning of heating surfaces which may make it desirable to consider more favorable arrangements than shown in Fig. 11. Often it is heard that the cleaning of flues is unnecessary, though it is well known that a very thin soot layer covering the heating surfaces reduces their effectiveness considerably.

Sometimes it is possible to increase the effectiveness of a given heating surface by gas rotation, which is self-explanatory, as by such means more hot gas particles come into intimate contact with the cooler heating surfaces thereby increasing the heat transmission by convection and conduction. Unquestionably there is also in the flues of each boiler a certain amount of heat transfer by gas radiation, but for the specific conditions in domestic boilers this influence is small.

As the details of the heating surface are important for the boiler's performance, it might be well to discuss this item still further. What is *Heating Surface?* According to general practice in the field of domestic boiler, heating surface, for instance in a steam boiler, is defined as the surface on the one side of which is the fire or fire gases, whereas on the other side, is the boiler water. With such a definition only the surface up to level L in Fig. 12 may be called heating surface; however, actual boiler tests disclose that as soon as the water nears the boiling point fine sprays of water wet the inside of the heating surface above the water line. It has been found that such surface, called dry heating surface, as compared to the wet heating surface below the water line, is at least as effective as the latter and sometimes even more valuable.

Table 1 shows certain boiler tests, where the water line in a steam boiler was raised from 47 in. to 57 in. and the wet heating surface thus considerably increased. Tests 1 and 2 refer to the pickup operation of the boiler with an average output of 1,830 sq ft of steam radiation and it will be noted that the efficiencies in both cases: namely, with low and high water line, were exactly the same. Tests 3 and 4 show a similar investigation but with steady operation of the boiler and again it is seen that the efficiencies are equal both for low and high water line settings. From these tests it can be concluded that dry and wet heating surface properly arranged are equivalent in their action.

The importance of the term heating surface has often led to the belief that a boiler's performance improves in direct proportion to the amount of its heating surface. In order to see somewhat clearer in this respect, note Table 2, in which there is a comparison covering a great many investigations and in which typical cases are shown.

The study in Table 2 is made for each of two boilers arranged in five groups, beginning with very large ones of about 800 sq ft heating surface, then large

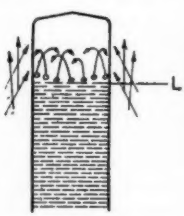
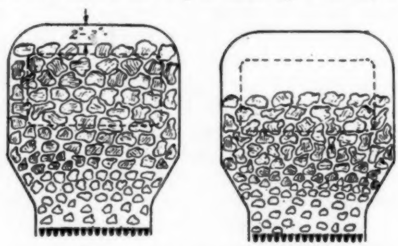


FIG. 12. WET AND DRY HEATING SURFACES



FIGS. 13 AND 14. VARIOUS FUEL HEIGHTS IN A COMBUSTION CHAMBER

boilers with about 400 sq ft heating surface, then medium boilers with about 200 sq ft heating surface, followed by small boilers of about 75 sq ft heating surface and finally comparing very small boilers with about 20 sq ft heating surface.

In the first group, boiler A has X sq ft heating surface, where boiler B has $X + 30$ per cent sq ft heating surface. The latter gave a maximum hourly output of only Y sq ft steam,[1] while boiler A with much less heating surface, delivered a maximum of $Y + 50$ per cent sq ft steam. Boiler A had a stack gas temperature of 780 F, and boiler B at highest output 580 F, both boilers working on equal chimneys, 30 in. x 36 in. in size and 95 ft high. It is interesting to note that the boiler with less heating surface and with higher stack gas temperature had an efficiency of 68 per cent for the maximum output, 74 per cent for $\frac{2}{3}$ and 78 per cent for $\frac{1}{3}$ of this load, while the corresponding figures for boiler B with 30 per cent more heating surface were 68 per cent, 73 per cent and 76 per cent. Boiler B with its lower stack gas temperature could not develop sufficient draft and hence was somewhat handicapped. In addition, it was found that the low draft was responsible for insufficient supply of secondary air so that boiler B with its large amount of heating surface showed an average CO value of 2 per cent whereas boiler A produced only 0.4 per cent

[1] 1 sq ft steam = 240 Btu/hour.

average CO, and it is known that each 1 per cent increase in CO reduces the efficiency 4 to 5 per cent. In conclusion, it is to be said that boiler A is a better boiler than B which has 30 per cent more heating surface, a 200 F lower stack gas temperature and 23 per cent more weight.

Similar results are disclosed in the other groups as can be seen from Table 2. In order to save time note the smallest products compared. Boiler K has a certain amount of heating surface X, whereas boiler L has $X + 17$ per cent heating surface, yet boiler L shows 14 per cent less maximum output. It was found in this case that the surplus heating surface was ineffectively arranged so that the average stack gas temperature of the boiler L at maximum output was 900 F, while boiler K with less heating surface and more output gave 700 F stack gas temperature. Both boilers had the same weight. They were connected to chimney 8 x 8-in. and 35-ft. height. However boiler K had at average load 74.5 per cent efficiency against 72.2 per cent for boiler L and at 50 per cent of the average load, the former realized 70.0 against 64.9 per cent.

It will be noted therefore that it is not correct to classify, rate and sell domestic cast iron boilers according to their heating surface. It may further be concluded that the judgment of boilers based upon their stack gas temperatures leads to erroneous results, especially if the influence of the products of combustion (CO_2 and CO) is neglected.

In this latter respect, the following facts found in actual tests may prove to be of interest. One boiler A was first fired according to Fig. 13 with the fuel crowded into the fire-box up to a few inches from the crownsheet, which is customary in laboratory investigations. After this the same boiler was filled to the middle of the fire-door only, as depicted in Fig. 14.

The diagram Fig. 15 refers to the first mentioned method of firing up to the crownsheet and gives:

an average CO_2 content of 13.9 per cent
an average CO of 0.6 per cent
and an efficiency of 70.0 per cent

In contrast to this diagram Fig. 16 shows the condition when the same boiler was filled only to the middle of the fire-door.

It indicates:

an average CO_2 content of 13.9 per cent
an average CO content of 0.2 per cent
and an efficiency of 72.5 per cent

which is easy to understand if it is recalled that the increase of 1 per cent in the CO content of the stack gases reduces the efficiency from 4 to 5 per cent.

Indeed these facts should be most carefully considered and the author is convinced that in so doing one will arrive at a new, or rather old item, namely, the Useful Service of a boiler. Regarding this Useful Service the writer would like to summarize his experience briefly:

1. A boiler must be able to deliver in the morning the necessary output easily and quickly, also under the most exacting and severest conditions of rapidly changing weather.
2. A boiler with too much heating surface and therefore a too low stack gas temperature will, even with laboratory efficiencies of 80 per cent, be unsatisfactory if its chimney cannot develop the draft necessary to obtain quickly the required high output.

TABLE 2. COMPARISON OF CHARACTERISTICS OF LARGE AND SMALL BOILERS IN RESPECT TO OUTPUT, HEATING SURFACES, WEIGHT, EFFICIENCY, ETC.

Boiler							Efficiency (%) for			
Max. H. S. Sq. Ft.	Name	Actual H. S. Sq. Ft.	Max. Output Sq. Ft. Steam 1 Sq. Ft. = 240 Btu/Hr	Av. Stack Temp. Max. Output F	Chimney Data Area = Sq. In. Height = Ft.	Weight Lb.	Max. Output	2/3 of Max. Output	1/3 of Max. Output	Remarks
800	A B	X X + 30%	X + 50% Y	780 580	30 x 36 95 Ft.	Z Z + 23%	68 68	74 73	78 76	Steam—Anthracite Average CO %...0.4 Max. Output.... 2.0

B 30% more H.S. 23% more weight. 1% CO = 4 to 5% in Efficiency.
A Better efficiencies, through higher stack gas temperature. 50% more reserve capacity in output.

							Max.	2/3	½	
400	C D	X X + 78%	Equal	815 595	20 x 20 60 Ft.	Z Z + 30%	71 71	76 74	78 75	Steam—Anthracite Average CO %...0.2 Max. Output.....1.0

D 78% more H.S. 30% more weight.
C Slightly better efficiencies, though much higher stack gas temperature. Same maximum output.

							Max.	2/3	½	
200	E F	X X + 40%	Equal	570 680	12 x 16 45 Ft.	Z Z + 15%	78 78	82 82	81 80	Steam—Anthracite Average CO %...0.1 Max. Output....0.15

F 40% more H. S. 15% more weight.
E Equal efficiencies—same maximum output.

TABLE 2. (*Continued*).

							Max.	2/3	½	
100	G H	X X + 35%	Equal	510 520	12 x 12 45 Ft.	Z Z + 12%	80.5 79	80.5 81	80 78	Steam—Coke Average CO at Max. Output—equal

H 35% more H.S. 12% more weight.
G Efficiencies at maximum and ½ load slightly higher. Same maximum output.

							Av. Load	½ Av. Load	
25	K L	X X + 17%	Y Sq. Ft. Water 1 Sq. Ft. = 150 Btu/Hr Y − 14%	705 910	8 x 8 35 Ft.	Same Weight	74.5 72.3	70.0 64.9	Water—Coke Average CO at Max. Output—equal

L 17% more H.S. 14% less maximum output.
K Efficiencies considerably better.
H.S. = Heating Surface.

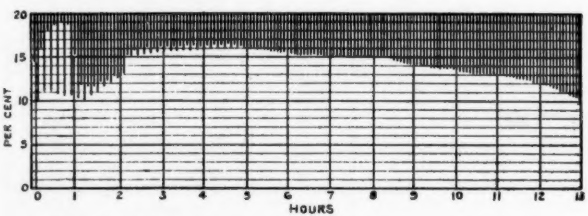


FIG. 15. MONO DIAGRAM WITH HIGH BOILER FILLING
(CO_2, 13.9%; CO, 0.6%; *e*, 70.0%)

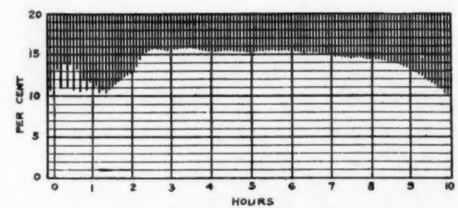


FIG. 16. MONO DIAGRAM WITH BOILER FILLING TO MIDDLE OF FIRE-DOOR
(CO_2, 13.90%; CO, 0.20%; *e*, 72.5%)

3. A boiler must be able to do such emergency work even with the poor chimney conditions frequently encountered in actual practice.
4. Replacement of a highly efficient boiler by one which has 2 or 3 per cent less efficiency is not only justified very often but works in the direction of higher "Useful Service," as it gives easily and quickly everything that the consumer wants and with satisfactory fuel consumption.
5. The economical operation of a house heating plant is less influenced by the boiler's efficiency during severe weather than during the many weeks when the system works at one-half or one-third, or even less of its rated capacity.
6. During those times correctly selected boilers need attention only at long intervals despite the fact that the combustion chamber is never filled up to

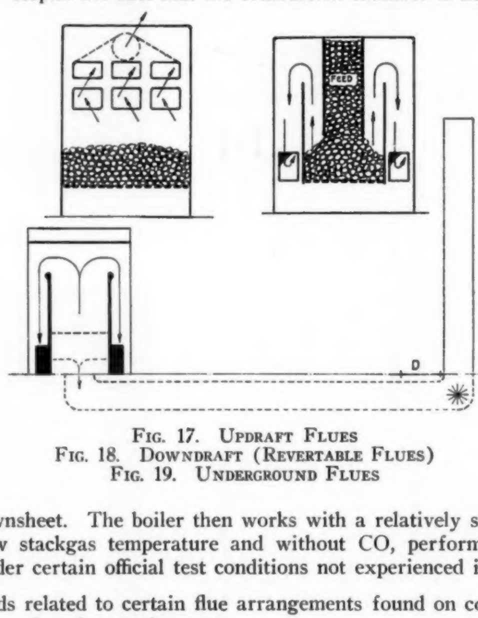


FIG. 17. UPDRAFT FLUES
FIG. 18. DOWNDRAFT (REVERTABLE FLUES)
FIG. 19. UNDERGROUND FLUES

the crownsheet. The boiler then works with a relatively shallow fuelbed, with low stackgas temperature and without CO, performing differently than under certain official test conditions not experienced in the field.

A few words related to certain flue arrangements found on coal boilers here and abroad may be of some interest.

Boiler flues are either arranged according to Fig. 17 where the gases continue their way upward from the fuel bed to the chimney or they can be arranged according to Fig. 18 where the gases have to pass through a down or revertable flue. Boilers of the latter type often have a large amount of heating surface and show low stack gas temperatures and high efficiencies in the laboratory.

However these boilers are sometimes difficult to start if a bypass is not provided, which connects directly for a certain time combustion chamber and chimney, and which offers, especially in relatively warm and damp weather and with poor chimneys, great conveniences. How important such a device is may be learned from publications in Europe which reveal the following facts:

A boiler with revertable flues (Fig. 19) was installed and connected with an

under floor breeching to a distant chimney. When first fired in mild Fall weather or after longer recesses during an advanced Spring the kindling wood, on top of which was a layer of coal, began to smoke considerably. This is quite understandable as the chimney which was wet due to rain, could not develop any draft on account of the very low stack gas temperature then prevailing.

To remedy this condition the janitor opened the cleanout D in order to start a paper fire at the bottom of the chimney and in so doing was overcome by the products of incomplete combustion present at that point.

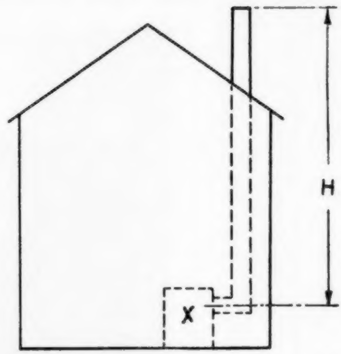


FIG. 20. HOUSE, BOILER AND CHIMNEY

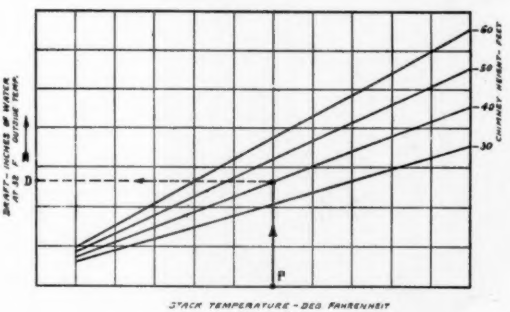


FIG. 21. DRAFT DIAGRAM

Many more important items could be brought up for discussion as for instance; detrimental action of gas leakage between boiler sections, beneficial influence of iron to iron joints and accurate machining, short circuiting of gases through the boiler, water capacity at water-line, disengaging areas and steam dome volume, foaming and priming, fluttering of flames in airtight oil boilers, steam water mixtures and other items as: application of pressure or suction in boilers, buckwheat and soft coal boilers, smokeless combustion and so on. However, as time does not permit discussion in such matters, certain suggestions which we think are of more general interest, will now be made.

Fig. 20 shows in diagrammatical form a house equipped with a given chimney of certain dimensions and a definite height H. Boiler X is brought

into this house and with the chimney provided must deliver a requested maximum amount of heat units. Following out this plan, Table 3 has been compiled from practical observations and shows chimney sizes and heights as a function of the boiler's maximum output.

The diagram Fig. 21 has been developed in which the theoretical draft is shown for a given outside temperature (of for instance 32 F) as a function of the chimney's height and of the stack gas temperature at the boiler's outlet.

This draft however is theoretical and cannot be obtained in actual practice because many items reduce its value, especially the cooling effect and leakage

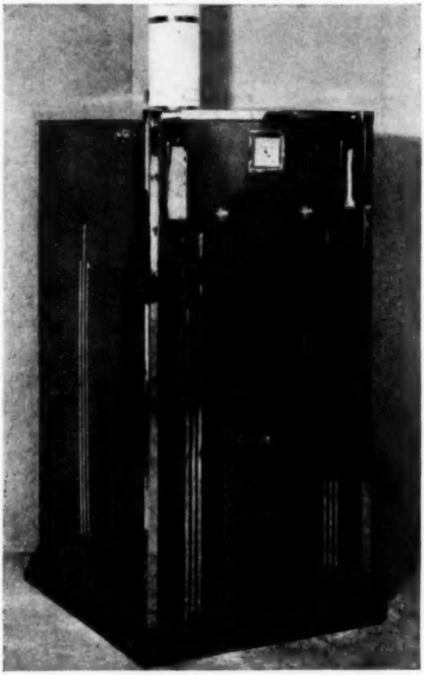


FIG. 22. MODERN BOILER DESIGN

of the chimney. One can then go into statistical search and find that for practical conditions only a certain fraction of the theoretical values can be safely counted upon. With such a practical chart which could be developed there would be only one more thing necessary to solve the problem completely. This one thing is a table such as Table 4 in which the boiler manufacturer gives among other data, the following six items:

1. Outputs.
2. Stack gas temperatures.
3. Chimney sizes and heights.
4. Draft requirements.
5. Efficiencies.
6. Time available fuel will last.

TABLE 3. CHIMNEY TABLE

Anthracite Coal

Maximum Boiler Output, Sq. Ft.		Chimney	
Steam	Water	Size, Inches	Height, Feet
250 to 500	350 to 800	8 x 8	30
500 to 1000	800 to 1600	8 x 12	35
1000 to 1500	1600 to 2400	12 x 12	35 to 40
1500 to 2500	2400 to 4000	12 x 16	40 to 45
2500 to 3500	4000 to 5600	16 x 16	40 to 50
3500 to 4500	5600 to 7200	16 x 20	50 to 55
4500 to 6500	7200 to 10000	20 x 20	50 to 60
6500 to 9500	10000 to 15000	20 x 24	60 to 75
9500 to 12500	15000 to 20000	24 x 28	75 to 85
12500 to 15000	20000 to 24000	30 x 30	85 to 100
15000 to 20000	24000 to 32000	30 x 36	90 to 115
20000 to 25000	32000 to 40000	36 x 36	110 to 125

TABLE 4. BOILER SELECTING TABLE

BOILER NUMBER	OUTPUT IN (1000's) B.T.U. PER HR. / OUTPUT—STEAM RADIATION SQ. FT.	90 / 375	102 / 425	114 / 475	126 / 525	138 / 575	150 / 625	162 / 675	174 / 725	186 / 775	198 / 825	210 / 875	222 / 925	234 / 975	246 / 1025	270 / 1125	294 / 1225	318 / 1325
X-5-4	ATTENTION PERIOD HRS.	19.7	17.7	15.8	14.4	13.3	12.3	11.4	10.6	9.8	9.3	8.7	8.3					
GRATE AREA	EFFICIENCY %	77.0	77.5	78.0	78.0	78.0	78.5	78.5	78.5	78.5	78.0	78.0	78.0					
SQ. FT.	STACK GAS TEMP. °F	305	320	335	350	365	380	395	410	425	440	455	470					
FUEL CAP.	GRATE LOAD PER SQ. FT./HR. LBS.	2.7	3.0	3.3	3.6	4.0	4.4	4.8	5.1	5.5	5.8	6.1	6.4					
LBS.	DRAFT REQUIRED IN. WATER	.027	.032	.037	.045	.052	.058	.062	.068	.075	.085	.095	.105					
ATTENTION	CHIMNEY SIZE INCHES	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12					
FUEL-LBS.	CHIMNEY HEIGHT FEET	35	35	35	35	35	35	35	35	35	35	35	35					
X-5-5	ATTENTION PERIOD HRS.			18.0	17.6	16.1	14.8	13.7	12.7	11.8	11.2	10.5	10.0	9.5	9.0	8.5		
GRATE AREA	EFFICIENCY %			78.0	78.0	78.5	78.5	78.5	79.0	79.0	79.0	79.0	79.0	78.5	78.5	78.5		
SQ. FT.	STACK GAS TEMP °F			305	320	335	350	365	380	395	410	425	440	455	470	485		
FUEL CAP.	GRATE LOAD PER SQ. FT./HR. LBS.			2.6	2.9	3.2	3.5	3.8	4.1	4.3	4.6	4.9	5.2	5.4	5.7	6.2		
LBS.	DRAFT REQUIRED IN. WATER			.037	.043	.048	.053	.060	.067	.073	.078	.083	.087	.095	.107	.120		
ATTENTION	CHIMNEY SIZE INCHES			8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12		
FUEL-LBS.	CHIMNEY HEIGHT FEET			40	40	40	40	40	40	40	40	40	40	40	40	40		
X-5-6	ATTENTION PERIOD HRS.					18.5	17.4	16.3	15.3	14.3	13.3	12.6	11.8	11.3	10.7	10.0	9.1	8.3
GRATE AREA	EFFICIENCY %					79.0	79.0	79.0	79.5	79.5	79.5	79.5	79.5	79.5	79.5	79.0	79.0	79.0
SQ. FT.	STACK GAS TEMP °F					315	330	340	350	365	375	385	395	405	420	440	465	490
FUEL CAP.	GRATE LOAD PER SQ. FT./HR. LBS.					2.8	3.0	3.2	3.4	3.6	3.9	4.1	4.3	4.6	4.8	5.2	5.6	6.2
LBS.	DRAFT REQUIRED IN. WATER					.045	.047	.050	.053	.060	.067	.072	.077	.082	.087	.097	.110	.123
ATTENTION	CHIMNEY SIZE INCHES					8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12	8x12
FUEL-LBS.	CHIMNEY HEIGHT FEET					40	40	40	40	40	40	40	40	40	40	40	40	40

With such information the procedure is self-evident.

(a) Determine maximum boiler load.

(b) Follow the vertical column at that load until you find the boiler you like for its efficiency, for its stack gas temperature, for its attendance period, or for any other reason.

(c) Check, if for that boiler, the chimney data coincide with the standard chimney table.

(d) Note the stack gas temperature and chimney height and check if the requested draft is actually available as taken from the draft diagrams.

(e) Under otherwise equal boilers, that one is the best which shows for one-half or one-third of its rated capacity most favorable characteristics for practical use.

Looking backward over the years consider what was known of boiler design and what progress has been made in scientific research and in practical manu-

facture. Let us not forget one point of great importance, namely, the presentation of our engineering products to the public. Formerly thrown into a dark part of the cellar the boilers indispensable for comfort and health, present themselves today as items of specific attraction (Fig. 22).

To review the items just discussed in connection with boiler design does it not appear as though the former standards of boiler selection; weight, grate area, heating surface, etc. must give way to the requirements of actual practice, namely, requested amount of heat and good efficiency under given draft conditions.

DISCUSSION

DR. C. W. BRABBEE (WRITTEN): Since the January meeting of our Society the Report of the Sub-Committee on boiler output has been published in the May issue of the "Official Bulletin" of the Heating, Piping and Air Conditioning Contractors National Association, and in it Mr. Harry Hart makes the following interesting comments on the author's suggestions:

"The paper also deals with the subject of firepot design which we will discuss in this report.

"As previously stated in the report, there seems to be a tendency on the part of some manufacturers to deviate from recognized desirable firepot design in order to gain a higher Net Load Recommendation by widening the firepot at the grate line. This is especially true in the small size of boilers.

"In order to discourage this practice we have decided to base our recommendations for the small size sectional boilers on the width of the firepot at a point 9 in. above the grate line instead of at the grate line."

W. H. CARRIER: I think the two pertinent facts in considering any boiler code have been brought out very well by Dr. Brabbée. I have always opposed, not because of any personal reasons (because I am not interested in this in any way personally), but from sound engineering grounds, the theory that the only criterion of boiler performance is grate area or the surface area. Progress can only be allowed to take place when you are freed from arbitrary restrictions and rule of thumb methods such as these. It must be based on actual performance under a given condition. I believe that any code that disregards this principle is injurious to the progress of the industry.

W. L. FLEISHER: I regret that H. M. Hart is not here. He could probably answer these points that Dr. Brabbée has brought up. Some years ago I was on the Boiler Code Committee, both of this Society and of the *Heating and Piping Contractors National Association.* In fact, I served on those committees for many years. The manufacturer who is being hurt by this grate area method of rating boilers was himself responsible for its arising. The necessity for tests and a large number of tests to determine the rating of boilers was found too expensive by the majority of boiler manufacturers at that time and in order to get some standard an arbitrary method of employing the grate area was used. It is not proper to blame either the arbitrary rules or the people that made them. The manufacturer himself was as much at fault as anybody. I regret that Mr. Hart, who was largely responsible for a good bit of this procedure, is not present to defend the *Association* to which he belongs.

DR. C. W. BRABBEE: We do not blame anybody, not even ourselves.

What I presented were certain facts showing that some research work concerning boiler ratings has been completed. I further explained how this new method could be practically applied.

In general, I believe if progress is to be made in our art we should try to make use of new findings; otherwise we will stand still instead of marching forward to new goals.

No. 990

SEMI-ANNUAL MEETING, 1934

WITH members and guests in attendance from 17 states, the District of Columbia, and Canada, totalling nearly 300, the 1934 Semi-Annual Meeting of the Society was held at The Inn, Buck Hill Falls, Pa., June 19-22. In addition to the usual technical sessions, an innovation was introduced in the form of non-technical addresses by nationally known men in the construction field and these discussions relating to the rehabilitation of the construction industry were enthusiastically received.

One of the three technical sessions was held jointly with the members of the *American Society of Refrigerating Engineers* and four papers were presented and discussed by the members of both societies.

Amendments to the Society's Constitution were introduced and discussed, then favorable action was taken on two changes in the By-Laws previously mailed to members of the Society.

Registration began on Tuesday morning, June 19, and during the day meetings of the Guide Publication Committee and the Committee on Research were held. In the evening members of the Council attended a dinner and meeting at which 1935 meeting places were selected, five members were nominated for service on the Committee on Research and other routine business was acted upon.

Pres. C. V. Haynes presided at the opening session on Wednesday, June 20, 9:30 a. m., and responded to the greetings of W. P. Culbert, president of the Philadelphia Chapter, who expressed the pleasure of the members acting as hosts for the large attendance.

At the conclusion of the technical program, President Haynes said that the Society had suffered a grievous loss and requested the members to stand with bowed heads as a tribute to the late Dean F. Paul Anderson. J. I. Lyle presented the following resolution in memory of Dean Anderson, past president of the Society and former Dean of the University of Kentucky College of Engineering, who died on April 8, 1934:

Whereas, God in his wisdom has seen fit to take from our midst our beloved past president, Dean F. Paul Anderson,

Whereas, the Society has been immeasurably benefited by his wise counsel and unfailing endeavors to further its best interests,

Whereas, his organizing ability and leadership in promoting the work of the research laboratory was outstanding, and

Whereas, his accomplishments in the academic world reflected credit to his University and this Society,

Whereas, his most genial personality, genuine comradeship and real sympathy for others endeared him to all our members and his associates.

Therefore, Be It Resolved, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS hereby expresses its sense of bereavement and irreparable loss; and

Be It Further Resolved, that this resolution be inscribed in the proceedings of the Society and a copy be sent to his family.

A motion by W. T. Jones, seconded by J. D. Cassell, that this resolution be adopted was passed by vote of the meeting.

Before closing the session, President Haynes announced that at luncheon immediately following, Society members and guests would have the pleasure of hearing Col. Willard Chevalier, of McGraw-Hill Publishing Co., who would speak on Examining the Construction Recovery Program.

W. H. Driscoll presided at the Get-Together Luncheon in the main dining room and presented the guest and speaker. Colonel Willard Chevalier, speaking from wide experience and understanding, analyzed the economic problems facing the construction industry and foresaw a definite trend toward better business. A fluent and forceful speaker, Colonel Chevalier was enthusiastically received by the A. S. H. V. E. group.

A joint meeting with the *American Society of Refrigerating Engineers* occurred on Thursday, June 21, 9:30 a. m., and was presided over by W. H. Carrier, past president of both organizations.

In opening the session Mr. Carrier greeted the members and welcomed the *A. S. R. E.;* whose interests in air conditioning are closely identified with the Society, and presented the following officers of both societies who were present: Dr. A. R. Stevenson, Jr., Schenectady, N. Y., president of the *A. S. R. E.,* and Harry Harrison, New Brunswick, N. J., vice-president; C. V. Haynes, Philadelphia, Pa., president of the A. S. H. V. E., John Howatt, Chicago, Ill., 1st vice-president, G. L. Larson, Madison, Wis., 2nd vice-president, and D. S. Boyden, Boston, Mass., treasurer.

A telegram of greetings from the Institution of Heating and Ventilating Engineers, Great Britain, was read, after which the authors of technical papers were presented.

The *American Society of Refrigerating Engineers* was responsible for two technical presentations and the first paper by Paul Bancel, New York, N. Y., discussed the subject of The Water Vapor Centrifugal Compressor.

D. W. McLenegan, a member of both organizations, was introduced by Mr. Carrier and described Electric Rate Making for Small Air Conditioning Units.

A five minute intermission was declared and President Haynes took the chair. He expressed pleasure at having Thomas S. Holden, vice-president in charge of statistics and research of F. W. Dodge Corp., as a guest and speaker. Mr. Holden, who is nationally known in the architectural and building construction field and who holds many civic offices, was presented by President Haynes and spoke on The Reemployment of the Construction Industry.

At the conclusion of Mr. Holden's talk, Mr. Carrier resumed the chair and the A. S. H. V. E. papers on the program were presented.

The closing period of the Semi-Annual Meeting 1934 of the Society, Friday, June 22, 9:30 a. m., was declared in session by Pres. C. V. Haynes, who

announced that the registration of members, guests and ladies was unusually large for a summer meeting and he expressed his appreciation for the attendance.

Report of Committee on Revision of Constitution and By-Laws

The first item of business was a report on amendments to the Constitution and By-Laws, which was presented by W. T. Jones. The proposed changes in the Constitution which were endorsed by the Council, were as follows:

Article C-II, *Section* 7. A Student Member shall be a person over sixteen (16) years of age who is regularly attending courses in an engineering college or school.

To Be Amended as Follows: A Student Member shall be a person *between the ages of 20 and 25 years,* who is regularly attending courses in an engineering college or *technical* school *at the time of applying for membership.*

On motion of W. T. Jones, seconded by J. D. Cassell, it was

Voted: that the proposed amendment be accepted and sent out to the membership for ballot in the prescribed manner.

Article C-III, *Section* 4. When a Student Member discontinues his regular studies in an engineering college or school, it shall be incumbent upon him to apply within one (1) year for advancement to either Junior, Associate or Member grade.

To Be Amended as Follows: When a Student Member discontinues his regular studies in an engineering college or technical school, it shall be incumbent upon him to apply within one (1) year for advancement to either Junior or Associate grade. *In no case shall the period of Student Membership exceed four (4) years.*

On motion of W. T. Jones, seconded by R. H. Carpenter, it was

Voted: that the proposed amendment be approved by the meeting and be sent out to the membership for ballot in the prescribed manner.

The following amendments to the By-Laws sent to the membership in advance of the Semi-Annual Meeting were read by Mr. Jones:

Article B-IV, *Section* 1. The Admission Fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1935 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

To Be Amended as Follows: The Admission Fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until *1940* and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

On motion of W. T. Jones, seconded by R. H. Carpenter, it was

Voted: that the amendment be adopted.

Article B-IV, *Section* 2. The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1935 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members three dollars ($3.00).

To Be Amended as Follows: The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until *1940* and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members *five dollars ($5.00)*.

On motion of W. T. Jones, seconded by A. W. Luck, it was

Voted: that the amendment be adopted.

Mr. Jones explained that an amendment to Article B-VIII—Committees, *Section* 1, had been proposed during the 40th Annual Meeting of the Society in New York and had been referred to the Committee on Constitution and By-Laws. The committee had considered the matter and had not approved of the proposal to change the present Article and permit appointment of other than Council members on the Executive, Finance, Membership and Meetings Committees as it felt that these committees should be chosen from the Council whose members are elected by the Society.

On motion of W. T. Jones, seconded by R. H. Carpenter, it was

Voted: that the proposed amendment to Article B-VIII, Section 1 of the By-Laws be rejected.

A new *Section* 11, of Article B-VIII, prepared by the Committee on Constitution and By-Laws and amended by Council is proposed as follows:

Section 11. At the first meeting of the Council after the Annual Meeting, or as soon thereafter as possible, the president shall appoint a Committee of at least five (5) members, whose duty it shall be to arrange for the publication of The Guide. This Committee shall be responsible for the text section and shall arrange for the inclusion of such new chapters as will furnish information regarding new developments.

In order that the wording of the various Sections of Article B-VIII may be in agreement, the Council suggested a change in the wording of the present *Section* 11, which would become *Section* 12:

Section 11. At the organization meeting of the Council, each year, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This Committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

To Be Amended as Follows: Section 12. At the first meeting of the Council after the Annual Meeting or as soon thereafter as possible, the president shall appoint a Committee of five (5) members to be known as the F. Paul Anderson Award Committee. This committee shall function according to the terms of the F. Paul Anderson Award. (See Appendix B.)

On Motion of W. T. Jones, seconded by J. D. Cassell, it was

Voted: that these amendments to be By-Laws be sent to the members of the Society in the prescribed manner.

Resolutions

Action was taken on the adoption of the following resolutions, which were presented by R. H. Carpenter, Chairman of the Resolutions Committee:

Resolved, that this Semi-Annual Meeting extend to the Philadelphia Chapter and its Committee of Arrangements who planned the enjoyable entertainment and other recreational features, its appreciation and thanks for the extremely pleasant time enjoyed by all.

Resolved, that the A. S. H. V. E. assembled at the Semi-Annual Meeting 1934, extend to the management and the staff of The Inn its hearty thanks and appreciation for the splendid accommodations and courteous services rendered to its members and guests.

Resolved, that the A. S. H. V. E. members attending the 1934 Semi-Annual Meeting express to the Program Committee and the speakers and authors of papers, their satisfaction and pleasure for the interesting and instructive data presented during the technical sessions.

Resolved, that the A. S. H. V. E. express its cordial thanks to the *A. S. R. E.* for its friendly cooperation and participation in the joint sessions and social events of the two organizations.

Resolved, that the 1934 Semi-Annual Meeting of the A. S. H. V. E. express its appreciation and thanks to the trade and daily press for their attendance and the generous publicity given to our meeting.

Resolved, that the members have observed with much pleasure the attendance of 10 of our past presidents and desire to express our hearty thanks for the continued interest of these "old timers"; and

Be It Further Resolved, that President Haynes receive the universal commendation of those in attendance, especially the ladies, for his fatherly interest in their welfare and enjoyment.

On motion of W. T. Jones, seconded by Albert Buenger, the resolutions were unanimously adopted, and the meeting adjourned.

PROGRAM SEMI-ANNUAL MEETING, 1934

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

THE INN, BUCK HILL FALLS, PA.

June 19-22, 1934

(All Events on Daylight Saving Time)

Tuesday—June 19

9:00 A.M. Registration.
9:30 A.M. Guide Publication Committee Meeting, W. L. Fleisher, *Chairman.*
10:30 A.M. Consulting Engineers Committee, Henry C. Meyer, Jr., *Chairman.*
1:30 P.M. Meeting of the Committee on Research, John Howatt, *Chairman.*
7:00 P.M. Dinner and Meeting of the Council.

Wednesday—June 20

9:00 A.M. Registration.
9:30 A.M. Greeting—C. V. Haynes, *President.*
Technical Papers—
Criteria for Industrial Exhaust Systems, by J. J. Bloomfield
Dry-Bulb vs. Effective Temperature Control, by A. E. Beals
Influence of Stack Effect on the Heat Loss from Tall Buildings, by Prof. Axel Marin
Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet

Thursday—June 21

8:30 A.M. Breakfast Meeting of Nominating Committee, W. R. Eichberg, *Chairman*
9:30 A.M. Joint Session with *American Society of Refrigerating Engineers,* W. H. Carrier, *Chairman,* A. R. Stevenson, Pres. *A. S. R. E.,* C. V. Haynes, Pres. A. S. H. V. E.
A. S. R. E. Technical Papers—
The Water Vapor Centrifugal Compressor, by Paul Bancel
Electric Rate Making for Small Air Conditioning Units, by D. W. McLenegan
10:30 A.M. Intermission
Address: The Reemployment of the Construction Industry, by Thomas S. Holden, Vice-Pres., F. W. Dodge Co.

A. S. H. V. E. Technical Papers—
Operating Results of an Air Conditioning System Compared with Design Figures, by J. R. Hertzler
What Is the Cooling Load Factor in Air Conditioning, by John Everetts, Jr.

Friday—June 22

9:30 A.M. Amendments to Constitution and By-Laws
Technical Papers—
Insulating Value of Bright Metallic Surfaces, by Prof. F. B. Rowley
Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by Prof. G. L. Tuve and C. A. McKeeman
Factors Affecting the Heat Output of Convectors, by Prof. A. P. Kratz, M. K. Fahnestock and E. L. Broderick

Entertainment

Tuesday—June 19

8:30 P.M. Concert, Ernest Gamble Trio.

Wednesday—June 20

12:30 P.M. Get-together Luncheon (*Main Dining Room*)
Address: Examining the Construction Recovery Program, by Col. Willard Chevalier, of McGraw-Hill Publishing Co., Inc.
1:30 P.M. Men's Golf Tournament—Buck Hill Course
2:00 P.M. Ladies Swimming Party at the Pool
7:00 P.M. Beefsteak Dinner at Skytop with *A. S. R. E.* Members and Ladies

Thursday—June 21

Morning: Morning Setups for Ladies—Bowling on Green or Swimming in the Pool
1:30 P.M. Bridge Luncheon for Ladies at Buck Hill Tennis Club
Research Cup Tournament for A. S. H. V. E. Members at Buck Hill Course
7:45 P.M. Banquet, Entertainment and Dance (*Main Dining Room*)

Friday—June 22

Morning: Automobile Trip for Ladies of A. S. H. V. E. and *A. S. R. E.* to points of interest in the Poconos
12:30 P.M. Pocono Mountain Trout Luncheon—(*Main Dining Room*)
Afternoon: Trip for Ladies to Skytop for Duplicate Bridge and Refreshments
2:00 P.M. Golf for Men at Skytop Course

COMMITTEE ON ARRANGEMENTS

R. C. BOLSINGER, *General Chairman*

Banquet Committee: M. F. Blankin, *Chairman;* J. D. Cassell and A. E. Kriebel.

Finance Committee: W. F. Smith.

Transportation and Publicity Committee: J. H. Hucker, *Chairman;* A. E. Dambly, H. H. Erickson, H. P. Gant and R. E. Jones.

Entertainment Committee: W. R. Eichberg, *Chairman.*

Registration and Reception Committee: H. G. Black, *Chairman;* Benjamin Adams, G. W. Barr, R. S. Arnold, J. D. Cassell, H. P. Gant, R. E. Jones, F. D. Mensing and A. J. Nesbitt.

Golf Committee: W. P. Culbert, *Chairman;* M. F. Blankin and Mrs. W. R. Eichberg.

Ladies Committee: M. C. Gillett, *Chairman;* Mrs. M. F. Blankin, Mrs. W. P. Culbert, Mrs. W. R. Eichberg, Mrs. H. H. Erickson, Mrs. J. H. Hucker and Mrs. A. E. Kriebel.

No. 991

CRITERIA FOR INDUSTRIAL EXHAUST SYSTEMS

By J. J. BLOOMFIELD* (*NON-MEMBER*), WASHINGTON, D. C.

KNOWLEDGE of the requirements of industrial exhaust systems, for the effective removal from the air of toxic dusts, fumes, gases and vapors, in an effort to maintain hygienic conditions in workrooms, is of importance to the engineer. Although the installation and use of correctly designed exhaust systems are essential in the maintenance of hygienically safe conditions, they are not in themselves a complete solution of the entire problem. Exhaust ventilation devices, like any other machinery, require constant care if adequate protection is to be afforded to the industrial worker. Hence, it is essential that such systems be tested from time to time, in an effort to determine how effectively they are functioning and if they are being maintained properly. Such an evaluation requires the actual determination of those substances in the air which the system was designed to eliminate. The present paper treats in a general manner some of the methods and instruments which may be used in testing the efficiency of industrial exhaust systems.

The classification of dusts, fumes and smoke, on the basis of their physical properties given in Chapter 15 of THE A. S. H. V. E. GUIDE 1934 will not be amplified in this discussion except that dusts will be discussed from the viewpoint of their action on the body; namely, those which upon inhalation come in contact with the blood stream and exert a generalized toxic effect (compounds of lead, arsenic, cadmium, etc.), and dusts which produce a localized reaction, such as certain fibrosis-producing dusts exemplified by silica, asbestos, etc., which usually confine their deleterious action to the lungs.

THE SAMPLING AND ANALYSIS OF DUSTS

Fibrosis-producing Dusts

The abundant evidence at hand showing that the inhalation of certain industrial dusts is an important factor in the causation of pulmonary disease has emphasized the significance of the quantitative aspects of this problem. A knowledge of dust content of the industrial atmosphere is required not only for the purpose of determining the extent of the hazard involved in various

---

* Sanitary Engineer, Bureau of Public Health.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buck Hill Falls, Pa., June, 1934.

manufacturing processes, but is also useful in measuring the efficiency of protective devices which may be used in the elimination of the dust hazard.

The properties of a given dust which determine its capacity to produce pulmonary pathology are the nature of the dust, that is, its chemical and mineralogical composition, its particle size and finally the quantity of the dust dispersed in the atmosphere.

One of the outstanding results of the last 20 years of research in the field of dust inhalation is the demonstration of the fact that, in general, the degree of health hazard associated with the inhalation of any dust, all other factors remaining constant, is dependent upon the mineralogical composition of the dust. For example, it is now established that the inhalation of certain types of dust, such as granite dust,[1] will in time produce fibrosis of the lungs, frequently associated with tuberculosis. In other cases exposure to dust may result in the production of a far lesser degree of fibrosis without subsequent tuberculosis; this is true of cement dust.[2] And finally, there are certain types of dusts which produce little lung fibrosis, as typified by marble dust.[3] In general it has been found that those dusts which are high in quartz content are the ones which most readily produce a disabling fibrosis of the lungs. Hence, the necessity for knowledge concerning the chemical and mineralogical composition of a dust is obvious.

So far as the size of the dust particles is concerned, it is apparent that in order for any given dust to produce injury to the lung it must gain access to the parenchyma of the lung, the site where the harmful effects of the dust take place. It is known that not all of the particles of inhaled dust gain access or are retained by the human lung.[4] For this reason it is of value to determine the size of the dust particles present in the industrial atmosphere.

With reference to the quantity of dust present in the air of a workroom it is apparent that when the dust concentration is high the exposed person will inhale a greater quantity in a given period of time than he will when the dust concentration of the atmosphere is relatively low, and since the rate of production of the disease is partially dependent upon the total amount of dust inhaled, this latter fact plays an important role in determining the time of onset of the end result. The need for the evaluation of the quantity of dust in the industrial atmosphere is obvious.

Composition of Industrial Dusts

Present day knowledge of the effects on the lungs of inhaled dust is more complete for quartz-containing dusts than for any other type of particulate matter. In general, it has been found that the harmfulness of a quartz-containing dust is in direct proportion to its quartz content. For this reason

[1] The Health of Workers in Dusty Trades. II. Exposure to Siliceous Dust (Granite Industry), A. E. Russell, R. H. Britten, L. R. Thompson, and J. J. Bloomfield. *Public Health Bulletin No* 187, July, 1929.

[2] The Health of Workers in Dusty Trades. I. Health of Workers in a Portland Cement Plant, L. R. Thompson, D. K. Brundage, A. E. Russell, and J. J. Bloomfield. *Public Health Bulletin No.* 176 (1928).

[3] Notes on the Effect of Inhaled Marble Dust as Observed in Vermont Marble Finishers, W. C. Dreessen. *Public Health Rep.*, 1934.

[4] Quantitative Measurements of the Inhalation, Retention, and Exhalation of Dusts and Fumes by Man, P. Drinker, R. M. Thomson, and J. L. Finn. *Journal Industrial Hygiene,* Vol. 10, No. 1, Jan., 1928.

in attempting to evaluate the harmfulness of a dust it is of the utmost importance to ascertain its quartz content. This is best done by a combined chemical and petrographic analysis.[5] It has been found in practice that samples of dust settled out of the atmosphere at the breathing level of the worker serve admirably for mineralogical analyses, which are best carried out by a competent geologist. Since no two dusts offer the same problem it is difficult to specify general procedure for such an analysis. Suffice it to say that each sample must first undergo a careful examination under the petrographic microscope, and in addition is further subjected to a complete chemical analysis with frequent petrographic examinations throughout the entire process. Only by such an analysis has it been found possible to determine accurately the percentage of quartz present in quartz-containing dusts. Table 1 presents the

TABLE 1. PERCENTAGE OF QUARTZ PRESENT IN VARIOUS INDUSTRIAL DUSTS

	Percentage of Quartz
Rock drilling dust (bituminous coal mine)	54.0
Granite cutting dust	35.2
Rock drilling dust (anthracite coal mine)	31.0
Brass foundry dust	19.0
Dust from raw mills in cement plant	6.5
Slate mill dust (Vermont red slate)	3.0
Silverware polishing dust	1.7
Anthracite coal dust	1.5
Bituminous coal dust	1.2
Cement dust	1.0
Slate mill dust (Vermont green slate)	trace
Talc mill dust	none
Marble cutting dust	none

quartz content of dusts obtained in various industries in which the industrial dust problem has received some study.

Size of Dusts

It has been demonstrated that particles of a size greater than 10 to 12 microns in longest dimension are very seldom found in the lungs. This absence of larger particles is partly due to the fact that the numbers of such particles greater than 10 microns in size present in industrial air is, as compared with the lower sizes, comparatively small and due to gravity and the protective action of the mucous surfaces of the upper respiratory tract, these larger particles do not penetrate to the terminal portions of the respiratory tract. Hence attention need only be given to those dust particles which are less than 10 microns in longest dimension.

In order to ascertain the size distribution of an industrial dust it is necessary to make particle-size studies of the dust under consideration. In practice the samples for such studies may be obtained by the use of the Owens Jet Dust

[5] The Quantitative Determination of Quartz (Free Silica) in Dusts, A. Knopf. *Pub. Health Rep.*, Vol. 48, p. 183, 1933.

Counter.[6] This apparatus projects the atmospheric dust directly on a microscope cover-slip. This cover-slip may then be properly mounted and examined microscopically. The particle size dimensions which are obtained from the microscopic study, are grouped in classes according to size, from which a percentage distribution curve is easily obtained. Fig. 1 presents the particle-size distribution of talc dust from the air of a workroom in which talc was being ground to a very fine state of subdivision. The measurements, from which the data for this figure were obtained, were made by means of an occular filar micrometer at a magnification of 1,000 diameters.[7] With this magnification it is possible to measure particles as small as 0.5 microns in diameter, while particles smaller than 0.5 microns are easily distinguished at this magnification and although not measured their presence is recorded.

Quantity of Dust

As pointed out earlier a knowledge of the quantity of dust dispersed in the industrial atmosphere is very important, since with any given dust the rate

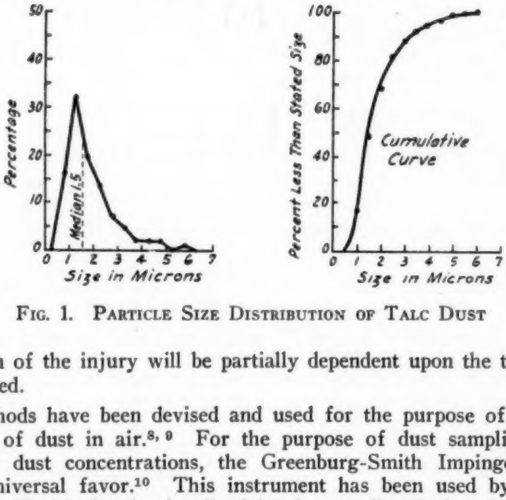


FIG. 1. PARTICLE SIZE DISTRIBUTION OF TALC DUST

of production of the injury will be partially dependent upon the total quantity of dust inhaled.

Many methods have been devised and used for the purpose of determining the quantity of dust in air.[8, 9] For the purpose of dust sampling in either high or low dust concentrations, the Greenburg-Smith Impinger apparatus now finds universal favor.[10] This instrument has been used by the United States Public Health Service in all of its dust studies during the past eleven years and is also being used by other workers in this country and abroad.

In this instrument, the air to be sampled is drawn through a glass tube and impinged at a high velocity on the flat bottom of a glass flask containing water or other suitable fluid. The dust is momentarily arrested, wetted by the collecting fluid, and in this manner trapped.

[6] Jet Dust Counting Apparatus, J. S. Owens. *Journal Industrial Hygiene*, p. 522, April 1923.
[7] E. M. Chamot and C. W. Mason. *Handbook of Chemical Microscopy*. 1932, p. 402.
[8] Dust Determinations in Air and Gases, E. R. Knowles. A.S.H.V.E. TRANSACTIONS, Vol. 25, 1919.
[9] Studies on the Industrial Dust Problem, Leonard Greenburg. *Public Health Rep.*, Vol. 40, No. 16, April 17, 1925.
[10] The Impinger Dust Sampling Apparatus as Used by the United States Public Health Service, Leonard Greenburg and J. J. Bloomfield. *Pub. Health Rep.*, Vol. 47, No. 12, Mar. 18, 1932.

The impinger apparatus consists essentially of two portions; *first,* a source of sufficient suction to draw the air to be sampled through the sampling device; and *second,* the sampling device or impinger itself, which consists of a container and the impinger tube. As a source of suction one may use either an electrically-driven pump or a compressed air ejector device.[10, 11] Fig. 2 depicts the essential portions of the impinger device, which consists of a straight piece of Pyrex glass tubing 15 millimeters in outside diameter and approximately 275 millimeters in length. The tube is drawn down in stream line form at its

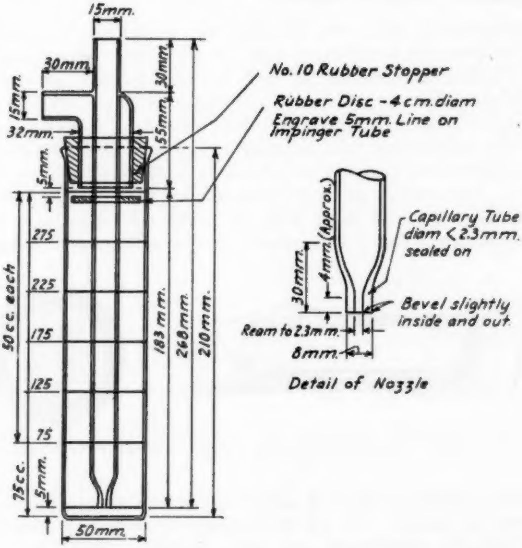


FIG. 2. MODIFIED IMPINGER ASSEMBLY AND DETAIL OF NOZZLE, WITH SPECIFCATIONS FOR CONSTRUCTION

lower end, to a tip with a 2.3 millimeters orifice. In sampling this orifice is kept at distance of 5 millimeters from the bottom of the flask. The suction connection is combined with the inlet tube. The flask is 50 millimeters in diameter and 210 millimeters in height and requires a fluid (water) volume of only 75 cubic centimeters to give the proper depth of immersion to the nozzle. An entrainment trap in the form of a rubber ring prevents the possible loss of the liquid (and dust) with the outgoing air.

In sampling, the outlet or suction elbow of the sampling flask is connected with the source of suction by means of a suitable length (25 ft) of noncollapsible rubber tubing. The duration of the sampling period should be such as to yield a satisfactory suspension of dust for analysis and is thus

[11] Modified Form of the Greenburg-Smith Impinger for Field Use, with a Study of Its Operating Characteristics, T. Hatch, H. Warren, and P. Drinker. *Journal Industrial Hygiene,* Vol. 14, No. 8, Oct., 1932.

dependent on the concentration of dust in the atmosphere. Under the usual industrial conditions, samples of from 10 to 30 cu ft of air yield sufficient suspended dust for analysis. Since a sampling rate of 1 cfm is maintained, this will require a sampling period of from 10 to 30 min.

The collecting efficiency of the apparatus is dependent upon adherence to the previously cited impinger tube dimensions and the sampling rate of 1 cu ft of air per minute. Experimental tests of this instrument against finely-divided silica dust suspensions in air have consistently yielded efficiencies of 98 per cent at the specified sampling rate.[11, 12]

Since practically all dusts are, to some extent, soluble in water, it is good practice to analyze the samples as soon as possible. Such practice tends to prevent any undue flocculation as well as any solvent action on the dust particles. In the laboratory the dust suspension in the sampling fluid may be filtered through a 325-mesh screen and then diluted so that the number of dust particles in the microscope field is equal to approximately 50 to 75. Two or more 1-cubic centimeter portions are placed in Sedgwick-Rafter cells for counting (Fig. 3). The microscope is of the ordinary type provided with a suitable eyepiece and objective and fitted with an Abbé condenser. A Whipple disc eyepiece micrometer (Fig. 3) is placed in the microscope eyepiece and the

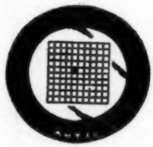


FIG. 3. (LEFT) SEDGWICK-RAFTER CELL. (RIGHT) WHIPPLE DISC

microscope tube length is adjusted so that the side of the ruling in the eyepiece is 1 millimeter in length. (A 7.5 X eyepiece, 16 millimeters objective and a tube length of 178 millimeters has been found to give this result.) An ordinary type of microscope lamp may be employed as a source of illumination. In order to provide a high degree of visibility for refractile objects it is best to lower the Abbé condenser system below the usual focusing point and to restrict the opening in the iris diaphragm. In making counts the microscope should be focused throughout the depth of the cell since some of the dust particles may remain in suspension. Since the counting cell is 1 millimeter deep and the area in the microscopic field is 1 square millimeter each count represents the amount of dust in a cubic millimeter of the sampling fluid. Knowing the original dilution of the sample and the number of cubic feet of air sampled it is an easy matter to compute the number of dust particles in the sample per cubic foot of air. It is of course necessary to make control dust counts on the sampling fluid.

Table 2 presents a summary of the average dust content of the air in certain dusty industries.

---

[12] Comparative Tests of Instruments for Determining Atmospheric Dusts, S. H. Katz, G. W. Smith, W. M. Myers, L. J. Trostel, M. Ingels, and L. Greenburg. *Public Health Bulletin No.* 144, 1925.

Poisonous Dusts

Some of the common poisonous dusts which exist in industrial atmospheres are the compounds of lead, cadmium, arsenic, mercury, etc. For the collection of these dusts one may employ any one of several instruments. For the collection of radioactive dusts,[13] the paper thimble has been successfully used. In

TABLE 2. AVERAGE DUST COUNT IN CERTAIN DUSTY TRADES

Industry	Dust Count in Millions of Particles per Cubic Foot of Air
SLATE FINISHING MILLS	
Floormen	1598.0
Loaders	1276.0
Disc crusher operators	312.8
TALC MINING	
Jack-hammer drillers	2159.8
Muckers	44.8
TALC FINISHING MILLS	
Crushers and cylindermen	14.0
Packers	50.1
MARBLE CARVERS	19.1
MARBLE CUTTERS	32.8
GRANITE QUARRYING	
Leyner drillers	144.4
Jack-hammer drillers	112.1
Plug drillers	36.9
CEMENT MILL, average of all operations	26.0
GRANITE CUTTING	
Hand pneumatic tool operatives	59.2
Machine pneumatic tool operatives	35.9
Attendant labor	17.0
ANTHRACITE COAL MINING	
Miners and miners' helpers	231.5
Attendant labor	31.1
BITUMINOUS COAL MINING	
Coal cutters and coal loaders	112.3
Attendant labor	3.9
SILVERWARE MANUFACTURING	
Dusty processes	5.2
Non-dusty processes	1.7
MUNICIPAL DUST (STREET CLEANERS)	
Congested district	4.1
Residential district	1.8
COTTON INDUSTRY	
Carding room	8.6
Weaving and spinning room	4.5

practice, a single thickness, Whatman extraction paper thimble, containing cotton wool, well fluffed out, is employed. Air is passed through such a thimble at the rate of 1 to 2 cfm, until sufficient dust has been collected, after which the contents of the thimble are subjected to a chemical analysis.

[13] Health Aspects of Radium Dial Painting. II. Occupational Environment, J. J. Bloomfield and F. L. Knowles. *Journal Industrial Hygiene*, Vol. 15, No. 5, Sept., 1933.

For precipitating dry, poorly conducting dust suspensions, non-rectified alternating current has been employed successfully.[14] In such a device air is passed though a glass precipitating tube at a fixed rate, using 15,000 volts, the dust being generally collected on a celluloid foil inserted in the tube. The material deposited on the foil may then be examined by chemical methods.

For most purposes, however, the impinger apparatus, already described in detail, is well adapted for the sampling of suspensions of poisonous dusts commonly encountered in industry. The collecting medium may be water, or some other suitable fluid. Blank tests on the sampling medium should be made.

In determining the extent of air pollution by poisonous dusts, one needs to consider two factors, irrespective of the sampling instrument employed. These are the need of a sensitive and reliable chemical method for the estimation of the quantity of the polluting material, and a sufficiently large sample of airborne dust must be collected to insure adequate amounts for the specific method of analysis employed.

THE SAMPLING AND ANALYSIS OF FUMES, GASES AND VAPORS

Fumes

The most common fumes encountered in industry are those originating from the oxidation of vapors formed by the heating of metals to high temperatures. Some of these fumes present in the air of workrooms are lead oxides, cadmium oxides, zinc oxides, etc. These fumes may be collected either by the electrical precipitating method[14] or by the impinger apparatus.[10] In case the latter instrument is used it is advisable that two impinger containers be employed in series. This precaution is necessary, since it has been found that the impinger device has a lower sampling efficiency for very finely divided materials, such as fumes, than for dusts.[11] Once the sample has been collected it may then be subjected to chemical analysis. The impinger apparatus has been used for the collection of chromic acid mist, the collecting medium employed being a normal solution of sodium hydroxide.[15]

Gases and Vapors

The gas and vapor hazards of modern industry are extremely important and the fatalities from poisoning by such substances increase year by year. Gases and vapors occur as raw materials, as products and as agents in manufacturing processes. Among some of the common gases one may list carbon monoxide, sulphur dioxide, hydrogen sulphide, hydrogen cyanide, ammonia, and some of the gases now used for refrigerating purposes, such as methyl and ethyl chloride. Many chemicals which are volatile at ordinary room temperatures, are used very extensively in industry in the form of solvents and lacquers. They constitute an extremely important group of contaminants and are characterized by their low density and easy volatility. Among some of the common vapors of a toxic nature found in industry are benzol, carbon disulphide, methyl alcohol, carbon tetrachloride and trichlorethylene.

---

[14] Alternating Current Precipitators for Sanitary Air Analysis. I. An Inexpensive Precipitator Unit, P. Drinker. *Journal Industrial Hygiene*, Vol. 14, p. 364, 1932.

[15] Health Hazards in Chromium Plating, J. J. Bloomfield and W. Blum. *Pub. Health Reps.*, Vol. 43, No. 36, Sept. 7, 1928.

Space does not permit the consideration of all of these gases and vapors. A discussion of one gas (CO), and one vapor (benzol) should suffice to illustrate the technic employed in determining the concentration of such substances in air.

There are several methods now available for the detection of carbon monoxide in air. Samples may be collected in glass containers by water displacement or by vacuum, after which they may be analyzed by either the Iodine Pentoxide method [16] or by the Pyrotannic acid method.[17] In the former method the gas is oxidized to carbon dioxide with the liberation of iodine, the amount of iodine being directly proportional to the amount of CO gas present in the sample. The iodine thus liberated is titrated with a standard sodium thiosulphate solution, from which the concentration of CO may be computed. In the pyrotannic acid method the reaction depends on the introduction of CO-free blood into the sampling container in order to form CO-hemoglobin. Pyrotannic acid, when introduced after the blood has been in intimate contact with the gas for 15 min or longer, will give a characteristic carmine red color, the intensity of which will depend on the amount of CO-hemoglobin present. Still another technic for the detection of CO gas in air, and one which is very rapid and sensitive, is the thermo-electric method using a catalyst.[18] The CO is oxidized to CO_2 in the presence of the catalyst with the liberation of heat. The quantity of heat which is liberated is measured directly on a millivoltmeter with the aid of thermocouples imbedded in the catalyst. Since the amount of heat liberated is directly proportional to the amount of CO gas present in the air, the millivoltmeter readings indicate the quantity of CO.

Benzol vapors may be detected in the air by the activated charcoal method, in which the vapors are absorbed by charcoal and the concentration determined by a computation from the increase in weight of the charcoal.[19] The interferometer [20] has also been used for the detection of benzol in air, and if no interfering vapors are present, this device is very sensitive and yields rapid readings. However, neither the charcoal nor interferometer method is specific for benzol in the presence of other hydrocarbon vapors. The method developed by Smyth,[21] which is a modification of a nitration method, has been found to be specific for benzol vapor in the presence of certain acetates and alcohols. In this method the air-vapor mixture is passed through an absorption device containing a mixture of equal volumes of sulphuric acid and fuming nitric acid. The benzene is converted to dinitrobenzene and then subjected to a chemical analysis in the laboratory.

---

[16] The Determination of Carbon Monoxide in Air Contaminated with Motor Exhaust Gas, M. C. Teague. *Journal Industrial Engineering Chemistry,* Vol. 12, No. 10, Oct., 1920.

[17] The Pyrotannic Acid Method for the Quantitative Determination of Carbon Monoxide in Blood and in Air, R. R. Sayers and W. P. Yant. *Techn. Paper* 375, Bur. Mines, 1927.

[18] Development and Characteristics of a Carbon Monoxide Recorder, S. H. Katz, D. A. Reynolds, H. W. Frevert, and J. J. Bloomfield. A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 211.

[19] Benzol Poisoning as an Industrial Hazard, L. Greenburg. *Public Health Reps.,* Vol. 41, No. 27, July 2, 1926.

[20] Application of the Interferometer to Gas Analysis, J. D. Edwards. *Tech. Paper No.* 131, Bur. Standards, 1919.

Ventilation of Heavier than Air Vapors, P. W. Gumaer. *Trans. National Safety Council,* 19th Congress, 1930.

[21] The Determination of Small Amounts of Benzene Vapors in Air, H. F. Smyth, Jr. *Journal Industrial Hygiene,* Vol. 11, No. 10, Dec., 1929.

APPLICATION OF TEST METHODS

The various methods and instruments for the sampling and analysis of dusts, fumes, gases and vapors have been presented. A few examples of the practical application of the technic described in the study of the efficiency of industrial exhaust systems, may serve to indicate the value of such criteria in the present problem.

As a result of a prolonged study of the health of the workers engaged in the various occupations of granite cutting,[1] it was possible to demonstrate that those persons engaged for many years in tasks associated with a dust exposure of less than 10 million particles per cubic foot of air were not suffering from silicosis or tuberculosis, the diseases most prevalent among these granite cutters. It was also possible to demonstrate that among these granite cutters the incidence of silicosis and tuberculosis, all other factors being equal, was directly proportional to the degree of dust exposure. The solution of the dust problem in the granite cutting industry, therefore, resolved itself in the removal of the dust at its source, to an amount less than 10 million particles per cubic foot,

TABLE 3. COMPARISON OF ATMOSPHERIC DUST CONDITIONS BETWEEN TWO GRANITE-CUTTING PLANTS

Occupation	Average Dust Count in Millions of Particles per Cubic Foot of Air; Winter Observations		
	Plants without Efficient Local Exhaust System	Plants with Efficient Local Exhaust System	
		Plant X	Plant Y
All pneumatic hand-tool operations	55.2	23.5	9.5
Surface cutting	45.0	15.3	10.6
Tool grinding	30.0	5.9	12.1
Sand blasting	6.9	3.5	5.5
General plant atmosphere	22.6	5.6	8.9

preferably by exhaust ventilation devices. Studies of the efficiency of such dust removal devices have been made using the dust determination at the worker's breathing zone as a final criterion.[22] In Table 3 a comparison of air dustiness is presented between granite cutting plants using exhaust ventilation devices and those not equipped with such protection.

It is apparent from the results presented in Table 3 that in plant X the exhaust devices in use with pneumatic tool operations needed attention, since the dust concentrations associated with these operations were slightly higher than the prescribed standard. On the other hand, the ventilation system in plant Y was apparently functioning satisfactorily at the time these studies were conducted.

In studying the degree of exhaust ventilation necessary to keep the dust at the worker's breathing level to an amount less than 10 million particles per cubic foot, the dust determination method again proved very useful.[22] Fig. 4

[22] A Study of the Efficiency of Dust-Removal Systems in Granite-Cutting Plants, J. J. Bloomfield. *Public Health Reps.*, Vol. 44, No. 42, Oct. 18, 1929.

presents the results of a study of the relation between the degree of air velocity at exhaust ducts and the amount of dust inhaled by granite cutters using various pneumatic tools. It is apparent from this figure that by maintaining an exhaust velocity of 1500 linear feet per minute at the face of the dust removal hood of the type investigated in the present study, the dust concentration at the worker's breathing zone will be less than 10 million particles per cubic foot.

The workers at the Harvard School of Public Health in conducting their studies on the removal of dust from granite cutting operations also used the

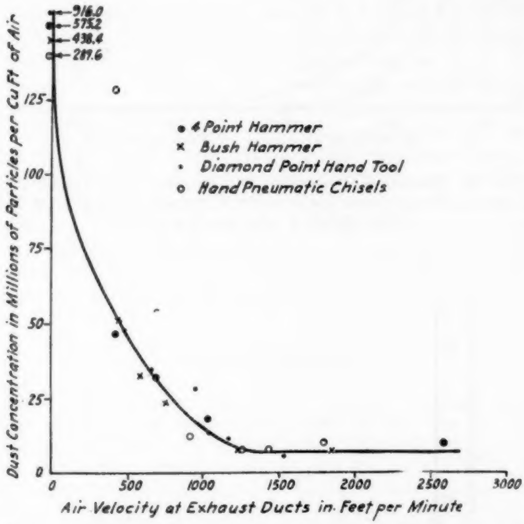


FIG. 4. GRAPH SHOWING THE RELATION BETWEEN THE DEGREE OF AIR VELOCITY AT EXHAUST DUCTS AND THE AMOUNT OF DUST INHALED BY GRANITE CUTTERS USING VARIOUS PNEUMATIC TOOLS

dust determination technic as a final criterion of the efficiency of the ventilation system designed by them.[23, 24] These investigators, as a result of their research, have been able to specify the correct design of local exhaust hoods and the minimum air flow requirements for the removal of dust generated by the use of pneumatic cutting tools in the granite cutting industry.

Mention has already been made of the use of the impinger apparatus in the determination of chromic acid mist in the air of chromium plating establishments.[15] These determinations were made at chromium plating tanks which were provided with exhaust ventilation. The degree of exhaust ventilation

---

[23] Control of the Silicosis Hazard in the Hard Rock Industry. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, T. Hatch, P. Drinker, and S. P. Choate. *Journal Industrial Hygiene*, Vol. 12, 1930.

[24] The Control of Industrial Dust, J. M. DallaValle. *Mechanical Engineering*, Vol. 55, No. 10, Oct., 1933.

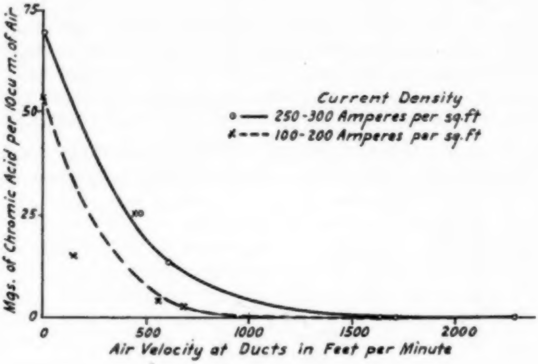


FIG. 5. RELATION BETWEEN AIR VELOCITY OF LOCAL EXHAUST SYSTEM AND AMOUNT OF CHROMIC ACID IN AIR FOR DIFFERENT CURRENT DENSITIES

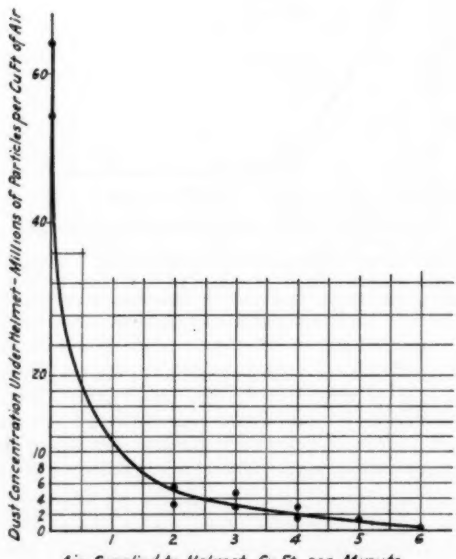


FIG. 6. CURVE SHOWING RELATIONSHIP BETWEEN THE VOLUME OF AIR SUPPLIED TO HELMETS AND THE NUMBER OF DUST PARTICLES IN AIR BREATHED BY WORKER

was determined at the same time that samples of air were obtained for chromic acid analysis. Such a study revealed the effectiveness of the exhaust system in removing chromic acid mist from the air. In Fig. 5 the relation between the degree of air velocity at the exhaust ducts and the amount of chromic acid in the air is shown. It is evident from this study that in order to keep the chromic acid content in the air to an amount less than 1 milligram in 10 cubic meters (the minimum amount found to cause no damage to the nasal septum), at least 1500 fpm of air movement at the face of the duct is necessary, especially at the higher current densities commonly encountered in electroplating.

Again, in studying the efficiency of sandblast helmets used in the protection of men working inside sandblast rooms, it was found that a relationship existed between the amount of air supplied to the helmet and the concentration of dust inside the helmet during blasting. In an attempt to determine the optimum air volume to be supplied to such protective devices, it was necessary to obtain dust samples from inside the helmet while varying the air volume, at the same time maintaining the dust concentration in the sandblast room (outside the helmet) constant. Fig. 6 shows the results of such a study and clearly indicates that the positive supply of 6 cu ft of dust-free air per minute will protect a worker under the operating conditions now in practice in sandblast rooms. The ultimate criterion of protection, however, is the result of dust determinations of the air within the helmet, that is, the air actually breathed by the worker and not the volume of air supplied.[25]

DISCUSSION

R. P. COOK: I am somewhat surprised at the author's success in his use of the Owens jet dust counter for determination of particle-size. My own experience with this instrument has not been entirely satisfactory due to a tendency of some of the small particles to pile up on the cover-slip to give the appearance of larger particles. This factor, it seems to me, is likely to result in considerable error both in particle-size and in number, especially at a magnification of 1,000 diameters. I am wondering if this same trouble is not experienced to some extent in the Greenburg-Smith impinger apparatus. It would seem to me that in this device the trouble might be either a building up of particles one on another, or a breaking down into smaller particles, depending upon the nature of the dust.

I suggest that the author add to his paper a brief description of the type and shape of the hoods in which he found that a face velocity of 1500 fpm or more is necessary. The shape of the hood, its location, and the volume of air handled are all of equal importance with the face velocity. The paper evidently refers to certain hoods described in the references and not to hoods in general.

N. D. ADAMS: It is now common knowledge that various forms of bacteria are carried about from place to place by different forms of dust, carried along by air currents. I should like to ask the author whether or not any bacterial counts were taken at the time of his dust counts as shown in Table 2.

J. J. BLOOMFIELD: There is no doubt that if attempts were made to count heavy deposits of dust taken with the Owens sampler, that one might encounter the difficulties suggested by Mr. Cook. But, at 1,000 magnifications, the field under examina-

---

[25] Sand and Metallic Abrasive Blasting as an Industrial Health Hazard, J. J. Bloomfield and L. Greenburg. *Journal Industrial Hygiene*, Vol. 15, No. 4, July, 1933.

tion is rather small and our experience has been that the dust particles are well scattered and readily measured. It has been suggested that both the Owens and the Impinger may fracture some of the dust but positive proof of such a phenomenon is lacking. We have never experienced trouble with flocculation of dust particles sampled with the Impinger.

It is realized that the size and shape of the hood, its location, and the volume of air handled are quite important in the effective removal of dust, and I have already called attention to the valuable work on this subject performed by Dalla Valle and Hatch at Harvard. It is not within the province of the present report to delve deeply into this phase of the work except to indicate that the criteria discussed in this paper will still need to be employed in judging the efficiency of the hoods designed and the volumes of air handled.

We have never made any bacterial counts in connection with our dust work, but recent studies on this subject by Wells and Drinker of Harvard and by certain research workers in Kansas indicate that the air-borne bacteria are of the harmless variety.

No. 992

DRY BULB VS. EFFECTIVE TEMPERATURE CONTROL

By A. E. Beals * (NON-MEMBER), Norwich, N. Y.

THE determination of effective temperatures and the location of human comfort zones [1, 2] upon the psychrometric chart have been recognized as notable scientific contributions and the Society is to be commended for sponsoring the long series of studies which have aided so materially in the rapid development of air conditioning for comfort. Well-merited appreciation is also due to those workers, whose skill and ingenuity has brought to such an orderly conclusion the mass of data and figures, which necessarily accrued during the years of investigation. All of the facts, figures and conclusions have been generously made public by the Society for unrestricted use by those interested in the progress of air conditioning.

In an early report we find an ingenious chart, devised to show at a glance the relations between dry- and wet-bulb temperatures and the resulting effective temperatures; also that experiment has clearly demonstrated that dry-bulb temperatures, per se, are wholly meaningless in terms of bodily sensations to heat and cold.

Of what practical use is all of this information and knowledge which has been collected and recorded so painstakingly? What are engineers doing with it?

A study of the comfort data shows that an effective temperature of 71 deg, within certain limits of relative humidity, is the summertime condition of maximum comfort [3] for 98 per cent of the subjects tested. Further, it was found that only about 50 per cent of the subjects felt comfortable at effective temperatures of 67½ deg on the cool side, and 75 deg on the warm side. It should be noted also that the subject must remain in and become inured to this environment. This acclimatization requires from two to three hours, depending upon the previous surroundings. Meanwhile a considerable degree of discomfort may be experienced from the contrast, particularly when the change is

* Consulting Engineer.

[1] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yaglou, A.S.H.V.E. Transactions, Vol. 29, 1923, p. 163.

[2] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou, A.S.H.V.E. Transactions, Vol. 29, 1923, p. 361.

[3] The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker, A.S.H.V.E. Transactions, Vol. 35, 1929, p. 269.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buck Hill Falls, Pa., June, 1934.

made from a higher to a lower level of temperature. Even when time is allowed for acclimatization, the range of comfort for the majority of people is exceedingly narrow when measured in terms of effective temperatures. Likewise, the permissible drop in effective temperature conditions is also exceedingly narrow if undue shock is to be avoided when changing from one to the other. In passing from a condition of high temperature or high humidity, and perhaps both, into a cooled room, an additional drop of only 1 deg of effective temperature will make an appreciable difference in the contrast and comfort experienced.

For instance, from a condition of 82 deg effective temperature, one will experience a much greater degree of temporary comfort, at an effective temperature of 76½ deg than at 74 deg if the relative humidity is at the proper point. In fact, an effective temperature of 76½ deg by contrast, would seem quite comfortable, if the relative humidity was such that perspiration was not unduly evidenced. Therefore, homes, expecting visitors, or offices, desiring business callers, should not be constantly maintained under a condition of maximum comfort, regardless of prevailing outside circumstances. This fact was rather belatedly recognized in the case of theaters, auditoriums and department stores, and a code [4] was devised governing the cooling of such places.

The A. S. H. V. E. Ventilation Standards stipulate that effective temperatures between 64 and 69 deg shall be maintained during the winter season, and between 69 and 73 deg during the summer when cooling is required. They also provide that the relative humidity shall not be less than 30 per cent nor greater than 60 per cent.

Considering only the summer conditions, this code covers quite a large area upon a psychrometric chart but the range of effective temperatures is small. As the conditions cited apply only to cases where the bodily sensation has reached equilibrium with the surrounding air, no consideration was given to outside temperatures and humidities, nor to the possible contrast in passing from one to the other. The need, however, for reducing the contrast between the outside and inside conditions was evident. Therefore, as a concession, it was further recommended that the inside dry-bulb temperature be maintained at 72 deg, plus one-third of the difference between the outside dry-bulb temperature and 70 deg.

In the light of all the information gained during the research into effective temperatures, it is a perfectly natural question to ask, why the outside dry-bulb temperature alone should, in any way, have a bearing upon the inside dry-bulb temperature. And a still more pertinent question would be, why should the outside *dry-bulb* temperature alone govern the inside *effective* temperatures.

If the dry-bulb temperature of itself is not an indication of bodily sensation on the inside of a theater, neither is it any criterion of the sensation experienced on the outside. Therefore, the relations of the dry-bulb temperatures within and without an auditorium, in no way measure the contrast experienced by the human body in passing from one to the other.

Nevertheless, this code completely ignores the adaptability and use of the principles of effective temperatures. It recommends that the control for inside conditions be based entirely upon the relation between the dry-bulb temperatures.

---

[4] A.S.H.V.E. Ventilation Standards, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383.

It merely stipulates that the inside effective temperatures be maintained within certain limits.

Still loyal to the old dry-bulb tradition, a table[4] based upon dry-bulb temperatures, giving the effective temperatures at different percentages of relative humidity, was published. Is that table of any particular benefit to the engineer, trying faithfully and conscientiously to operate his cooling plant to best advantage? If any attention whatever is to be paid to effective temperatures, this table might better have been based upon such temperatures, giving both dry- and wet-bulb values for the recommended inside effective temperatures and percentages of relative humidity. The engineer then would have a chance to at least check up and determine whether or not he was still within bounds.

To show the indefiniteness of this table, it is only necessary to examine the range of effective temperatures allotted to the dry-bulb temperature of 77 deg. These effective temperatures comprise practically the entire range from the lower to the upper allowable limits, the wet-bulb temperatures ranging from 57½ to 67 deg.

Also, according to the rule for auditoriums and the like, the inside temperature of 77 deg is that required for an outside temperature of 85 deg. But an outside dry-bulb temperature of 85 deg might be attended by any wet-bulb temperature between 64 and 80 deg, with a corresponding range of effective temperatures from 75 to 82 deg.

Imagine the contrast in going from an outside effective temperature of 82 deg into one of 69.7 deg! Furthermore, it is certain that one would feel much more comfortable outside at a temperature of 85 deg dry-bulb and 75 deg effective temperature than on the inside at 77 deg and an effective temperature of 72.6 deg.

Coincident with the adoption of the code by the Society a committee published an explanation of the use and application of the effective temperature chart.[5] This committee went even further than the code in repudiating the principle of effective temperatures. In its recommendations of conditions to be maintained indoors, for exposures of less than three hours, the use of effective temperatures was discarded altogether—inside as well as outside. It is the committee's suggestion that the inside dry-bulb temperatures vary in accordance with the same "rule of thumb" formula as expounded by the code, with the additional limitation that a constant inside dew-point of 57 deg be maintained. Thus, after a clear and concise exposition of the effective temperature chart and comfort zone, this committee casts aside its practical application in air-conditioning, and relegates this really monumental piece of research work to the status of an academic curiosity.

The constant dew-point of 57 deg for inside conditions was evidently chosen because, within the recommended range, a line, indicating the various conditions at this dew-point, lies well within the area of the comfort zone as defined by the code. But the resulting effective temperatures along this line are more in accord with permanent comfort, to which one must become acclimated, than with temporary comfort in contrast to the prevailing outside conditions.

[5] How to Use the Effective Temperature Index and Comfort Charts, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 410.

However, the maintenance of inside conditions at any constant dew-point is merely a pleasing fiction. In actual practice, with the usual methods of control now employed, this "just isn't done."

A thermostat, placed near the outlet of a washer, and set to maintain a constant wet-bulb temperature of the air leaving it, is incapable of maintaining a constant dew-point in an auditorium, subject to varying ratios of moisture to heat to be removed. For instance, assume that the dry-bulb thermostat is set to maintain a temperature of 72 deg in the room, and it is attempted to hold a constant dew-point of 57 deg. Also assume that a ratio of 1.37 grains of moisture per 1 Btu of heat has to be removed. To accomplish this, the saturated temperature of air leaving the washer would need to be 54 deg.

Suppose now that the load changes to such an extent that the ratio of moisture to heat removed becomes 0.75 grains per Btu. In order to maintain the 72 deg dry-bulb temperature at 57 deg dew-point in the room, it would then be necessary for the engineer to reset his thermostat for 55½ deg. If the outside temperature changes to 85 deg and the conscientious engineer consults his table, he finds that the inside temperature should be 76½ deg with 64 deg wet-bulb. Accordingly he immediately (?) resets his dry-bulb thermostat. Then he wonders what's going to happen.

As the inside temperature goes up, the ratio of moisture to heat removed also changes until it reaches 1.74 grains per Btu. To meet this demand, the wet-bulb thermostat again requires resetting for a saturated air temperature of 50¾ deg. But how is the engineer to know that?

By the time the harassed engineer thinks he is all set, maybe his inside load has changed again so that it is necessary to remove only one grain of moisture per Btu of heat. This requires a saturated air temperature leaving the washer of 54½ deg. And again, how is the engineer to know that?

Changes of the outside conditions and variations of the inside requirements are constantly taking place. The audiences in a theater are never there long enough to permit their bodily sensations to come into equilibrium with the environment. Under the rules and regulations promulgated by the code and the Society's committee, it is by accident, pure and simple, if a theater is ever found to be, for even a few fleeting moments, in a condition approximating temporary comfort. Many times, when one leaves, he wishes he had not gone in, and oftentimes, when one enters, he is tempted to back out, but to save the price of admission he decides to suffer and possibly contract a severe cold.

In actual operation of an air conditioning plant, instead of a constant dew-point of 57 deg within a theater, a constant wet-bulb temperature of the air leaving the washer is maintained. If the inside dry-bulb temperatures are varied with the outside dry-bulb temperatures, the ratio of moisture to heat required to be removed determines the dew-point, which varies accordingly. At the same time the wet-bulb and the effective temperatures vary also.

Another table, supplementing that of the code and setting forth "desirable indoor air conditions," has been published.[5] This table is based upon an inside, constant dew-point of 57 deg, and the inside dry-bulb temperatures are varied with the outside dry-bulb temperatures. The relations between these three temperatures have no bearing upon comfort sensations, and the effective temperature relations are uncontrolled.

In practice no attempt is made to maintain a constant, inside dew-point. In cases where the dry-bulb temperature recommendations are attempted, the wet-bulb and effective temperatures are entirely out of control of the cooling apparatus. If, however, a constant wet-bulb temperature of air leaving the washer is maintained at 50 deg, the comfort conditions established by this table will, under inside, full load requirements, be fairly well approximated. Under half-load requirements, the comfort conditions are not so good. The inside dew-points will vary from 51 to 60 deg for outside temperatures ranging from 90 to 100 deg; for outside temperatures from 74 to 80 deg, the dew-points will vary from about 52½ to 54½ deg. In the final analysis of its practical application to air conditioning, it is thus evident that this table merely recommends that inside *dry-bulb* temperatures be maintained in accordance with varying outside *dry-bulb* temperatures—and nothing more.

It is true that with a properly chosen wet-bulb temperature of air leaving the washer, the inside conditions can be held well within the established, average comfort zone. It is, however, a debatable question whether the conditions established by varying the dry-bulb temperatures are best either for the temporary comfort of the patrons or the financial interest of the theater owner. The costs for power, water and attendance, in the operation of this class of equipment, is one of the large factors in deciding whether the apparatus shall be installed or not. The temporary comfort of the patrons is a large factor in determining the amount of box office receipts. Therefore, the interests of all concerned will best be served when the maximum temporary comfort of the patrons can be assured at a reasonable cost of operation.

In public buildings, the criterion for inside temperature and humidity is the contrast experienced in passing from one condition to the other. This contrast is measured solely by the relation which the inside *effective* temperature bears to the outside *effective* temperature. The contrasting *dry-bulb* temperatures, taken by themselves, have no measuring nor comparative values whatever.

The wet-bulb temperature, prevailing on the outside, is a factor of just as much importance as the wet-bulb temperature, assumed to be maintained inside, or as either of the dry-bulb temperatures. As for the inside dew-point, that means nothing whatever.

Other influences, besides the actual temperature and humidity of the outside air, have an important bearing upon the degree of contrast experienced upon passing into a cooled room. Before entering a theater, the patrons have been exposed to the radiation from hot pavements and building walls, and the majority have indulged in more or less bodily activity. Each accentuates the sensation of warmth. After becoming seated, a period of relaxation ensues and all bodily activity practically ceases—unless involuntary shivering is produced due to too rapid withdrawal of heat. Therefore, for temporary comfort, too great a difference between the outside and inside effective temperatures should not be permitted. Neither should the relative humidity be so low that too rapid evaporation of perspiration is induced.

A number of years ago the writer saw an architect's specifications for a large auditorium which stipulated that the capacity of the air conditioning equipment should be sufficient to maintain, at all times, "an inside effective temperature midway between the outside effective temperature and an effective tempera-

ture of 66 deg." This, so far as the writer is informed, was the first and only recognition of the practical adaptability of the principle of effective temperatures. This proposal was made and the limiting temperature of 66 deg was chosen before later investigations had determined that 71 deg effective temperature is the condition of maximum summer comfort.

In the light of later investigations, an effective temperature midway between the outside effective temperature and an effective temperature of 71 deg would appear to be a reasonable compromise between the condition of maximum comfort and the discomfort of the outside environment. It would insure relief from the outside conditions and a sense of temporary comfort without undue shock, when passing either in or out.

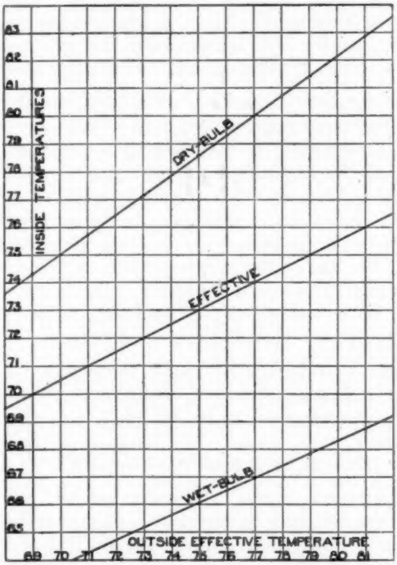


FIG. 1. SUGGESTED INSIDE CONDITIONS TO BE MAINTAINED CORRESPONDING TO VARIOUS OUTSIDE EFFECTIVE TEMPERATURES

The relative humidity should follow rather closely the "perspiration line."[6] This, through the required range, approximates 60 per cent for the lower and 50 per cent for the higher temperatures. Thus, while perspiration would not be unpleasantly evident, the rate of evaporation would not be sufficient to foster a sensation of chilliness. A regulation of this nature, which is based upon scientific principles governing inside conditions to be maintained, would seem

[6] Where Is the Perspiration Line?, by A. E. Beals, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1933.

to be ideal for short time exposures. It is also a regulation which is susceptible of 100 per cent automatic control.

With complete automatic control the human equation is entirely eliminated from the operating problem, and with the inside effective temperatures continually varying in accordance with the outside effective temperatures, a pleasing sensation of change will unconsciously be experienced. Also, regardless of sudden and wide variations of outside conditions, the inside environment will always be commensurate therewith, and extreme contrasts, in passing either in or out, will be avoided. The old time slogan of "Twenty Degrees Cooler Inside" has been entirely discredited.

Thus far only the comfort of patrons has been made the paramount consideration, but the trouble and expense to be borne by the owner in the rather

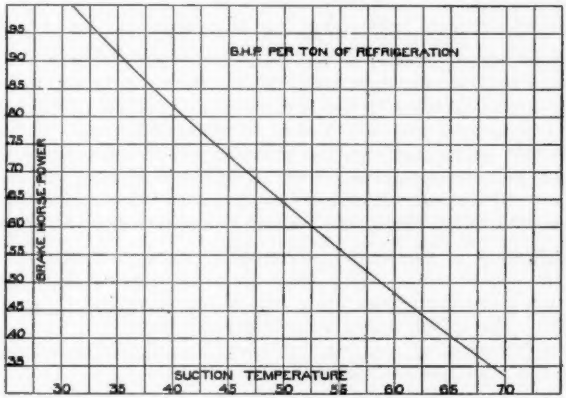


FIG. 2. BRAKE HORSEPOWER PER TON OF REFRIGERATION WHEN UNIT IS OPERATING UNDER VARIOUS SUCTION TEMPERATURES

complicated operation of air conditioning equipment is also entitled to consideration.

It is an easily ascertained fact that there is not a refrigerating unit used at present in connection with comfort cooling that is operating to good advantage. This is not the fault of the refrigerating equipment, which is usually of the best, nor is it the fault of the refrigerating engineers and designers. It is because of the requirements imposed by the air conditioning engineers.

One factor contributing to this state of affairs is the general practice of maintaining a constant wet-bulb temperature of the air leaving the washer. Another is the idea that it is better to intensively dehumidify a portion of the recirculated air. They are both at fault from the viewpoint of obtaining refrigerating economy. The first entails the maintenance of a constant temperature at which the refrigerating equipment must absorb heat. The second necessitates that this temperature be kept at the lowest point required by the maximum ratio of moisture to Btu of heat to be removed.

A refrigerating machine is an apparatus which absorbs heat at a lower temperature level and discharges it at a higher level. The high level of discharge is fixed within a relatively small range by the temperature of the available condensing water supply. Therefore, the lower the temperature at which the heat must be absorbed, the greater will be the power required. There are other factors which also influence the economical operation of the refrigerating unit, not necessary to discuss here.

By automatically controlling the inside effective temperatures in accordance with the outside effective temperatures, the maximum of temporary comfort can be provided for the patrons. By utilizing this same control to vary, in accordance with the requirements, the temperature at which the refrigerating machine must absorb its heat, a very considerable saving in costs of power and water can be effected. The average saving, over the present methods of operating, will at a conservative estimate amount to at least 30 per cent. As this saving is predicated upon maintaining a greater degree of temporary comfort, the best interests of both the patrons and the owner are thus assured.

The chart, Fig. 1, shows suggested inside conditions to be maintained, corresponding to various outside effective temperatures. This chart is based upon an inside effective temperature, which shall be midway between the momentary outside effective temperature and an effective temperature of 71 deg. The dry- and wet-bulb temperatures are so arranged that at an inside effective temperature of 71 deg, the relative humidity will be 55 per cent; at an inside effective temperature of 76 deg, the relative humidity will be 50 per cent.

As an indication of one source of saving in cost of operation, Fig. 2 is presented for consideration. This chart shows the approximate brake horse power per ton when the refrigerating unit is operating under various suction temperatures. The values shown are conservative because a 15 per cent heat loss from the refrigerating apparatus has been assumed, and the efficiency of the compressor has been figured at 85 per cent.

It is evident from Fig. 2 that the higher the temperature at which it is possible to utilize the cooling water in an air washer, the less will be the power cost.

DISCUSSION

F. C. HOUGHTEN: The paper by Mr. Beals is valuable and timely in that it points out limitations in the application of data from the A. S. H. V. E. Research Laboratory and elsewhere to modern air conditioning. The paper clearly shows the need for further clarification and development of summer air conditioning standards. In considering this subject it is well to keep in mind just how far our present standards are based upon laboratory research carried on by the Society and to what extent they are still based upon field observations.

The Research Laboratory worked on this subject from the time it was established until about 1927, publishing a number of reports giving the relation between a person's feeling of warmth and various combinations of temperature, humidity and air motion. Also, in 1923 the Laboratory published the results of its study of optimum effective temperatures for winter heating conditions. This much of the work was carried on by the Society's Laboratory in Pittsburgh. In 1929 Prof. C. P. Yaglou published the results of his findings of a study of optimum temperature conditions for definite periods of indoor exposure during the summer.

I believe there is little difference of opinion regarding the application of the laboratory's comfort data for winter heating conditions. There is, however, as pointed out by the author room for considerable difference of opinion regarding the application of the findings of the Laboratory and Professor Yaglou to summer cooling. This is, however, not due in any sense to error in the original findings, but rather to the fact that a person entering a cooled room in the summer has his body and clothing in equilibrium with outside atmospheric conditions, or with a high temperature. When he comes into the cooled space his body and clothing, wet with perspiration, give off heat to the new environment at a higher rate than required to maintain temperature equilibrium and as a result, he may experience a chill. As soon as his body becomes in equilibrium with the new environment, he will automatically desire atmospheric conditions corresponding to the laboratory findings of the Research Laboratory. Hence, two rather definite standards could easily be established for inside air conditions; one standard applying to a person entering the condition from a higher outside temperature, and another standard applying after he had become acclimated. The second standard should conform to the laboratory findings. The first set of standards, or the desired atmospheric condition for a person entering from a high outside temperature has never been studied by the Society's laboratory, but has been built up through practice, largely through box-office complaints from theater patrons. It is possible that an additional intensive laboratory study of the subject would be fruitful.

It should be pointed out that the committee report on How to Use the Effective Temperature Index and Comfort Charts,[5] referred to by the author of the paper, does not present additional laboratory findings, but was the result of a review of all of the previously published data on the subject, tempered with experience gained through air conditioning application. The limitations of relative humidity and dew-point contained in this report were not definitely based upon laboratory research but upon general experience and opinion.

F. C. McIntosh: Is it not true that our effective temperatures for summer weather were determined under indoor conditions? Outdoor effective temperatures might vary considerably from those determined, unless our tests used a strong proportion of radiant heat. A proper determination of outdoor effective temperatures for summer might be necessary for the idea, suggested by the author, of using them to determine the proper effective temperatures indoors.

A. E. Beals: As pointed out by Mr. Houghten, two definite standards can easily be established for desirable indoor air conditions; one applying to permanent comfort, and the other applying to temporary comfort. This was recognized by both the Code Committee and the Committee which published the report on How to Use the Effective Temperature Index and Comfort Charts.

Of these two standards, it would seem that the determination of conditions applying to temporary comfort is the more important. A determination of the conditions applying to permanent comfort is not of much consequence. The number of occupants in a condition of permanent comfort are, at the most, relatively few and they occupy this space day after day and for long durations of time. These occupants are not at all concerned with effective temperatures. It is only necessary for them to agree upon the indoor conditions most pleasing to their personal reactions and maintain that condition regardless of outside variations. They are not comparing the indoor condition, which they have determined to be most suited to their personal comfort, with any other condition.

On the other hand, persons passing from outside conditions into a cooled space, and remaining there for only short periods of time, are most vitally concerned with effective temperatures. It has been definitely determined that effective temperature is the only measure of bodily sensations of heat and cold. Persons entering a cooled

space from outside conditions for only short periods of time are comparing, through their bodily sensations, two entirely different atmospheric conditions. This comparison can not be measured by dry-bulb temperatures.

It is undoubtedly true, as Mr. McIntosh points out, that the heat sensation experienced outdoors will be higher than the effective temperature as registered by thermometers on account of the effect of radiant heat. It is also true that the heat sensation will be further modified by wind effect, but in the opposite direction. Therefore, the radiant heat factor will, to a certain extent, be neutralized by the cooling effect of the wind.

Another indeterminate factor is the effect of bodily exertion. This will, however, be subconsciously kept to a minimum. The net effect of the three outdoor factors, which will not be operative on the inside, will be to increase the sensation of outdoor heat above the effective temperature as registered by thermometers.

Since, for temporary comfort, a compromise from the condition of maximum comfort is desirable, it merely remains to determine approximately the extent of this compromise and by what standard it should be measured.

Heretofore the standard of comparison has been dry-bulb temperatures. The same modifying factors, which are brought forward as objectionable to the effective temperature standard, will apply to dry-bulb temperatures as a standard. Furthermore, wet-bulb temperatures, both outside and inside, are additional factors operating to still further modify the dry-bulb temperature standard.

Integrating wet-bulb temperatures with dry-bulb temperatures into effective temperatures eliminates the most important modifying factor inherent in the use of dry-bulb temperatures as a standard of comparison. It also provides the only standard of comparison between two atmospheric conditions to which bodily sensations are responsive.

No. 993

INFLUENCE OF STACK EFFECT ON THE HEAT LOSS IN TALL BUILDINGS

By AXEL MARIN *(*NON-MEMBER*), ANN ARBOR, MICH.

THE purpose of this paper is to present the results of investigations made in the Penobscot Building, Detroit, Mich., to determine the influence of stack effect on the heat loss from a tall building. The tests conducted in this building are the first in a series to be made in investigating this subject, and the results should not be interpreted as applying to any building until additional tests with other buildings have shown similar agreement between the calculated heat loss and the heat input as determined by test. The investigations are to be carried on during the next heating season at Detroit, and it is hoped when these tests are completed that the influence of stack effect on heat losses in tall buildings may be properly accounted for.

By *stack effect,* as applied to tall buildings, is meant the pressure difference existing between the inside and outside of a building due to the temperature difference. This pressure difference tends to increase the inflow of air on the lower floors of a building by adding to the wind effect and opposes the effect of the wind at the upper floors. For any given inside and outside temperature, the pressure difference at any elevation is theoretically proportional to the distance this place is from the neutral zone.[1]

The work is being done in cooperation with the Technical Advisory Committee on Heat Losses from Buildings of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It was made possible through the courtesy of the owners of the Penobscot Building and the building engineer, Mr. G. M. Lewis, who made available space for the study and assisted in every manner possible; and the Detroit Edison Co. who installed all the test equipment, made all the arrangements, and collected the test data.

THE BUILDING

The Penobscot Building, Fig. 1, is a modern 47-story office building located at the corner of Griswold and Fort Streets, Detroit, Mich. It is 565 ft from street level to roof, has a volume of 6,858,360 cu ft, and contains 73,940 sq ft of direct radiation and 19,250 sq ft of equivalent radiation in fan coils.

* Associate Professor of Mechanical Engineering, University of Michigan.

[1] The Neutral Zone in Ventilating, by J. E. Emswiler, A. S. H. V. E. TRANSACTIONS, Vol. 32, 1926.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1934, Buck Hill Falls, Pa., by J. H. Walker.

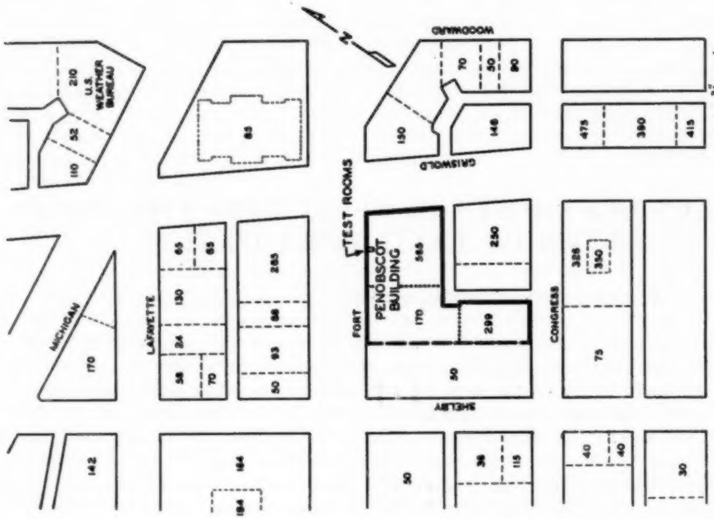


FIG. 2. PLAN SHOWING LOCATION OF PENOBSCOT BUILDING WITH RESPECT TO SURROUNDING BUILDINGS AND THEIR HEIGHTS

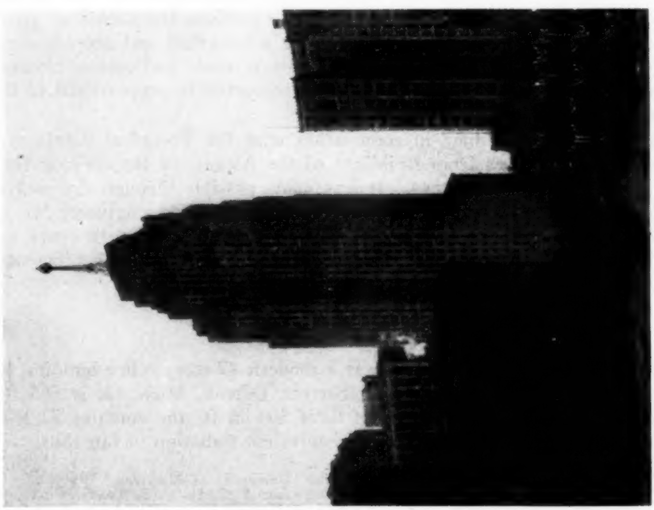


FIG. 1. PENOBSCOT BUILDING, DETROIT, MICH.

Fig. 2 shows a ground plan of the Penobscot Building and the other buildings in the immediate vicinity with their elevations indicated. Fig. 3 is a cross-section of the 8th floor which is the same for all floors up to the 31st floor. The cross-section decreases slightly from this point up to the top floor by set-backs at approximately every four floors.

General Plan

In planning the tests to determine the influence of stack effect on heat loss from a tall building, it seemed advisable to investigate at least three rooms in a building that have the same directional exposure and that are as nearly alike in every respect as possible with the exception of elevation.

Test Rooms

Three rooms on the north side of the Penobscot Building midway from the corners were used for the first series of tests—one on the 8th floor, one on the 21st floor, and the other on the 32nd floor. Figs. 4 and 5 show the test rooms on the 21st and 32nd floors with the testing equipment in place, and Fig. 6 is a plan view of the 8th floor test room with the details of the outside wall.

Table 1. Location, Exposure and Heat Loss of Test Rooms

Test Room	Exposed Wall Sq. Ft.	Exposed Glass Sq. Ft.	Lineal Ft. of Crack	Calculated Transmission Loss—Btu per Hour per Degree Temperature Difference	Distance from Ground to Mid-height of Test Rooms
8th Floor	57.6	29.6	26.67	48.53	105 ft. 6 in.
21st Floor	120.06	59.2	53.34	94.67	253 ft. 4 in.
32nd Floor	67.2	29.6	26.67	47.40	378 ft. 10 in.

Table 1 gives the area of the exposed wall and glass for each room, the calculated transmission loss in Btu per hour per degree temperature difference for wall and glass, the lineal feet of crack for the windows, and the elevation of the test floors above the ground.

Description of Apparatus and Tests

The amount of steam supplied to the radiators was determined by collecting and weighing the condensate. Special flat tanks were constructed for this purpose that could be disconnected quickly, the accumulated condensate weighed, and the tank connected again. The heat output of the radiators was manually controlled by throttling the radiator valve to maintain a constant temperature in the test room. The pressure of the steam supplied and the temperature of the condensate leaving the radiator were recorded, making it possible to determine the heat output of the radiator. Temperature of the air in the test room at 6-in. and 5-ft levels from the floor and 6 in. from the ceiling were recorded at 15 min intervals, as well as the temperatures of the outside air and the temperatures of the surrounding rooms.

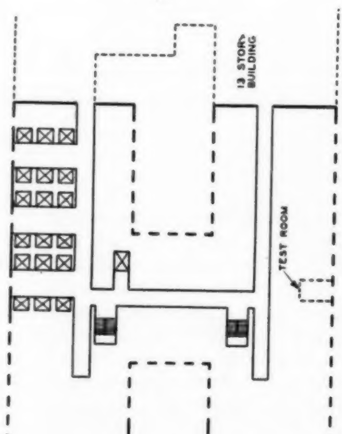


FIG. 3. CROSS-SECTION OF BUILDING AT 8TH FLOOR

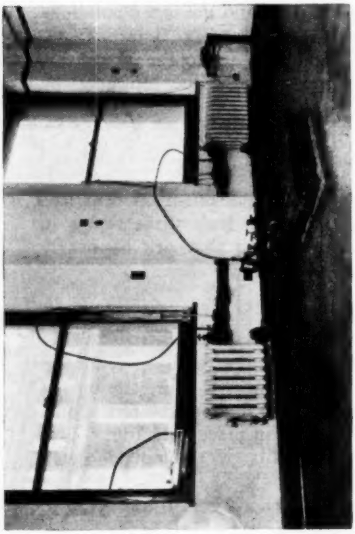


FIG. 4. TEST ROOM ON 21ST FLOOR

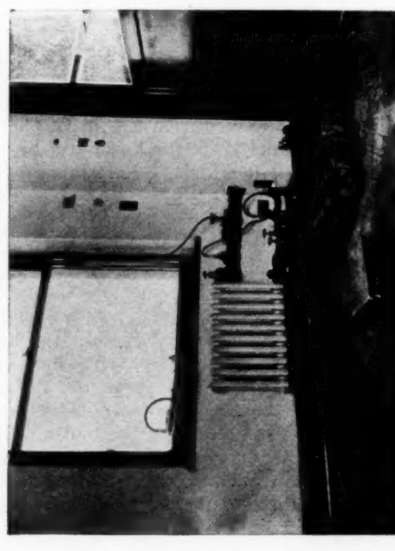


FIG. 5. TEST ROOM ON 32ND FLOOR

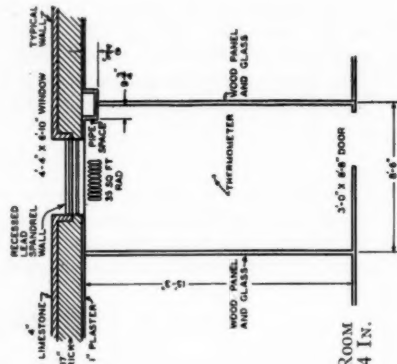


FIG. 6. PLAN OF TEST ROOM ON 8TH FLOOR (11 FT.-4 IN. CEILING)

The wind velocity and direction, the per cent of possible sunshine, the barometric pressure, and the outside temperature were obtained from the local weather bureau, which is located at the Majestic Building about three blocks away. The outside temperature obtained by thermometers suspended outside the windows at the various floors and 6 in. away from the wall surface were used in calculating the heat loss. These temperatures were practically identical with the temperatures reported by the weather bureau.

An attempt was made to measure the amount of infiltration or exfiltration from the test room by replacing the transom over the door with a tight fitting board that had a one square foot opening, and measuring the velocity at this place with an anemometer. The velocities in all cases were too low to move the anemometer wheel. The infiltration figures used in the calculations were based on the results of, Air Leakage Studies on Metal Windows, by Houghten and O'Connell [2] corresponding to the actual pressure drops measured across the windows. The pressure difference was measured with an inclined draft gage connected to copper tubing that extended through a hole drilled in the window sash.

RESULTS

The principal results are presented in Table 2. Column 5 gives the heat input in Btu per hour from the steam to maintain the test rooms at the temperatures indicated in Column 3 for the various outside temperatures listed in Column 14. The figures in Column 5 include the necessary corrections for any heat loss or gain from or to the surrounding rooms.

Column 6 is the heat input from steam corrected to a common inside temperature of 72.6 F. Column 6 is obtained by multiplying the values in Column 5 by the ratio of 72.6 F minus the outside temperature over the actual temperature difference between inside and outside. The temperature 72.6 F is the average room temperature for all tests.

Column 7 shows the amount of heat which would be necessary to maintain the test rooms on the various floors at 72.6 F, if the exposed wall and glass areas were the same as the 8th floor test room.

The calculated transmission loss, Column 8, is the sum of the quantities obtained by multiplying the area in square feet of exposed wall and that of glass by their respective transmission coefficient (U) and by the temperature difference between the air inside and outside. The temperature difference is 72.6 F minus the outside temperature.

The transmission coefficient (U) is

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{f_0} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3}}$$

in which

f_1 and f_0 = Surface coefficients.
k_1, k_2, etc. = The conductivity of the respective material.
x_1, x_2, etc. = The thickness of the various materials in inches.

The outside wall of the 8th floor test room has 38.1 sq ft of stone veneered wall and 19.5 sq ft of a recessed lead spandrel construction. The stone

[2] A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928, p. 321.

veneered wall is composed of the following material: 1 in. plaster, 17 in. brick, and 4 in. of cut limestone; and the recessed lead spandrel section is made up of 1 in. plaster, 8 in. brick, and a 2½ in. lead spandrel. The glass area for

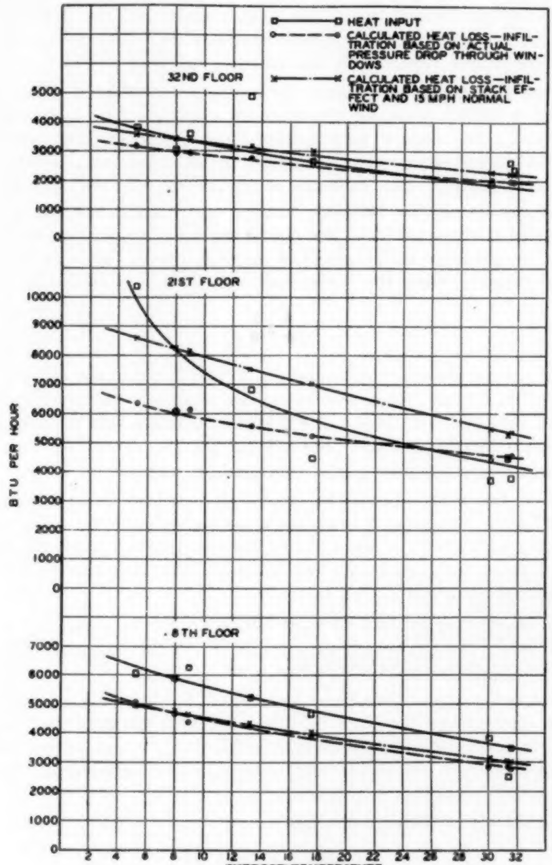


Fig. 7. Comparison of Calculated Heat Loss and Actual Heat Imput to Maintain Test Rooms at 72.6 F

the 8th floor room is 29.6 sq ft. The transmission coefficient for the stone veneered section, using values from The Guide 1934 is

$$U = \frac{1}{\frac{1}{1.65} + \frac{1}{6.0} + \frac{1}{3.3} + \frac{17}{5} + \frac{4}{12.5}} = 0.208 \text{ Btu per hour per sq ft per deg.}$$

and for the recessed spandrel wall

$$U = \frac{1}{\frac{1}{1.65} + \frac{1}{6.0} + \frac{1}{3.3} + \frac{8}{5} + \frac{2.5}{111}} = 0.370 \text{ Btu per hour per sq ft per deg.}$$

The total transmission loss in Btu per hour per degree temperature difference for the exposed wall and glass of the 8th floor test room is equal to

$$38.1 \times 0.208 + 19.5 \times 0.370 + 29.6 \times 1.13 = 48.53$$

To obtain the total transmission loss for any given temperature difference for the 8th floor room, 48.53 is multiplied by the temperature difference.

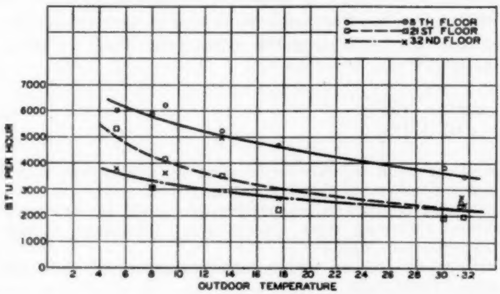


FIG. 8. BTU REQUIRED TO MAINTAIN TEST ROOMS AT 72.6 F REDUCED TO 8TH FLOOR EXPOSURE AS A BASE FOR THE PURPOSE OF COMPARISON

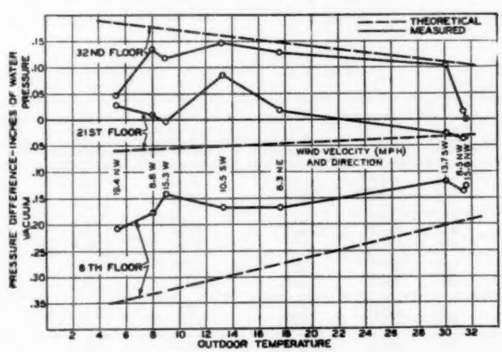


FIG. 9. COMPARISON OF MEASURED PRESSURE DIFFERENCE ACROSS WINDOWS WITH THEORETICAL DIFFERENCE DUE TO STACK EFFECT

The method employed in calculating the transmission loss for the 8th floor test room was also used in calculating the transmission loss for the 21st and

32nd floor test rooms. The 21st floor room has 102 sq ft of stone veneered wall, 18.06 sq ft of recessed spandrel wall, and 59.2 sq ft of glass. The outside wall of the 32nd floor test room has 67.2 sq ft of stone veneered wall and 29.6 sq ft of glass, with no recessed section.

The total calculated heat loss, which is the sum of the transmission loss and the infiltration loss, is shown with the infiltration loss calculated in two different ways. In one case the infiltration loss is based on the actual pressure drop across the window, and in the other case the infiltration loss is based on THE GUIDE method.

THE GUIDE method would be the one a designer would naturally use to determine the infiltration, when calculating the heat losses. This method assumes the neutral zone to be at mid-height, and determines the equivalent wind velocity, due to stack effect and wind that is responsible for infiltration. The expression used in calculating the equivalent wind velocity, taken from THE GUIDE 1934 is

$$M_e = \sqrt{M^2 - 1.75a}$$
$$M_e = \sqrt{M^2 + 175b}$$

where

M_e = Equivalent wind velocity to be used in conjunction with results of Air Leakage Studies on Metal Windows.

M = Wind velocity upon which infiltration would be determined if the temperature difference were disregarded. (Assumed in our computations as 15 miles per hour.)

a = Distance of windows under consideration from mid-height of building if above mid-height.

b = Distance if below mid-height.

Fig. 7 shows the variation in heat supply in Btu per hour to maintain the test rooms at a temperature of 72.6 F at the 5-ft level when the outside temperature varies from 5.3 F to 31.6 F. The calculated values of the heat losses,

TABLE 2. SUMMARY OF RESULTS

I	II	III	IV	V	VI	VII	VIII	IX	X
Date	Room Number	Inside Temperature at 5 ft level	Inside Relative Humidity in percent	Btu Input per Hour From Steam	Btu Input per Hour From Steam Corrected To 72.6F—Room Temp.	Btu Input per Hour From Steam Corrected To 72.6F—Room Temp. & 8th Floor Exposure	Btu Loss per Hour Through Walls and Glass (Calculated for 72.6F)	Btu Loss per Hour Due to Infiltration Based on Actual Pressure Drop Through Windows and 72.6F (Calculated)	Btu Loss per Ho Due to Infiltrati Based on Stack Ef & 15 mph Normal W (72.6F Inside)
1-29-34	853	70.2	20.0	5806.7	6030.0	6030.0	3266.0	1776.0	1668.0
	2153	72.5	21.0	10352.2	10352.2	5310.0	6371.0	-16.5	2220.0
	3250	71.5	22.5	3745.9	3745.9	3833.0	3190.0	-16.5	414.0
2-27-34	853	73.3	15.0	5958.5	5890.0	5890.0	3135.0	1541.0	1600.0
	2153	73.0	22.0	6069.7	6030.0	3090.0	6116.0	-7.0	2130.0
	3250	73.0	17.0	3015.2	2998.0	3070.0	3062.0	-63.6	397.0
1-30-34	853	69.2	23.0	5896.5	6225.0	6225.0	3087.0	1314.0	1575.0
	2153	72.2	20.0	8035.5	8080.0	4142.0	6021.0	100.8	2095.0
	3250	72.1	21.0	3557.9	3580.0	3670.0	3015.0	-51.5	391.0
3-20-34	853	73.1	17.5	5283.7	5220.0	5220.0	2878.0	1368.0	1470.0
	2153	72.8	21.0	6887.6	6800.0	3490.0	5614.0	-5.1	1958.0
	3250	72.9	18.5	4091.9	4065.0	4980.0	2811.0	-44.3	365.0
3-16-34	853	72.6	20.0	4666.6	4666.6	4666.6	2649.0	1268.0	1362.0
	2153	72.2	25.0	4448.3	4460.0	2288.0	5207.0	1.6	1815.0
	3250	72.6	20.0	2611.3	2611.3	2675.0	2607.0	-31.0	338.0
1-26-34	853	72.7	33.0	3807.2	3807.2	3807.2	2063.0	794.0	1043.0
	2153	72.1	29.0	3709.8	3710.0	1902.0	4023.0	425.0	1402.0
	3250	73.0	29.0	1921.1	1903.0	1948.0	2015.0	-21.4	261.0
3-14-34	853	73.1	24.0	2529.1	2498.0	2498.0	1999.0	835.0	1020.0
	2153	72.7	28.0	4545.4	4545.4	2330.0	3900.0	651.0	1360.0
	3250	72.9	24.0	2650.8	2632.0	2695.0	1953.0	-6.4	253.0
3-7-34	853	73.8	31.0	3585.3	3480.0	3480.0	1990.0	806.0	1015.0
	2153	73.3	28.0	3842.9	3795.0	1945.0	3881.0	581.0	1352.0
	3250	73.0	23.0	2389.1	2343.0	2400.0	1943.0	0	252.0

computed according to the methods previously explained, are also plotted against temperatures to show the relation between the calculated heat loss and outside temperature.

The test points of the heat supplied for the 8th floor test room, shown in Fig. 7, fall on a smooth curve with the exception of the 9 F and 31.4 F points. The 9 F point is high, due no doubt to the 15.3 mile per hour wind from the West, which is almost parallel to the outside wall. The only observed difference in outside conditions for the 9 F point and 8 F point, aside from the 1 deg temperature difference, is the wind velocity. This increased heat supply for the 9 F point may be due to the more rapid removal of the air film on the outside wall, thereby increasing the transmission loss. It does not seem possible to account for this increase as infiltration, because the pressure difference across the window is less for the 9 F test point than for the 8 F test point. It is interesting to note that the same sort of variation in heat supply for the 8 F and 9 F points is present in the curves of heat supply for the 21st and 32nd floor rooms. No reason is known as to why the 31.4 F point is low. The 31.6 F point and the 30.1 F point agree with the other points and they are more rational.

It is evident from the curves of calculated heat loss for the 8th floor room that there is little difference in the results obtained using the two methods of calculating the heat loss, but it is surprising to note that the curve of heat supply is approximately 20 per cent higher than the calculated results throughout the range of outside temperature investigated.

The test curve of heat supply for the 21st floor test room shown in Fig. 7 increases more rapidly in the region of the 5.3 F and 9 F test points than do the curves of heat supply for the other floors in this same temperature region. This increase may be due to the fact that the 21st floor is about at the neutral zone and the conditions of pressure are very unstable, varying from a slight vacuum to a slight pressure during many of the tests.

TABLE 2. (*Continued*)

XI	XII	XIII	XIV	XV	XVI	XVII	XVIII
Total Btu Loss per Hour Col.VIII plus Col.IX (Calculated)	Total Btu Loss per Hour Col.VIII plus Col.X (Calculated)	Actual Pressure Drop Through Windows Inches of Water	Outdoor Temperature (Observed)	Outdoor Relative Humidity in Percent (U.S.Weather Bureau)	Wind Velocity (mph) and Direction (U.S.Weather Bureau)	Percent of Possible Sunshine (U.S.Weather Bureau)	Barometric Pressure Inches of Hg (U.S.Weather Bureau)
5047.0 6354.5 3173.5	4934.0 8591.0 3604.0	-.208 +.028 +.046	5.3	60.0	19.4 NW	100	29.45
4676.0 6109.0 2998.4	4735.0 8246.0 3459.0	-.177 +.010 +.134	8.0	56.0	8.8 W	100	30.03
4401.0 6121.8 2963.5	4662.0 8116.0 3406.0	-.142 -.003 +.117	9.0	61.0	15.3 W	100	29.50
4246.0 5608.9 2766.7	4348.0 7572.0 3176.0	-.168 +.084 +.147	13.3	54.0	10.5 SW	100	29.51
3937.0 5208.6 2576.0	4031.0 7022.0 2945.0	-.168 +.018 +.128	17.6	48.0	8.3 NE	100	29.71
2857.0 4448.0 1993.6	3106.0 5425.0 2276.0	-.119 -.027 +.103	30.1	60.0	13.7 SW	75	29.41
2834.0 4551.0 1946.6	3019.0 5260.0 2206.0	-.139 -.038 +.013	31.4	64.0	8.5 NW	100	29.70
2796.0 4462.0 1943.0	3005.0 5233.0 2195.0	-.129 -.035 0	31.6	60.0	15.8 NW	60	29.43

The reason for the large difference in the two curves portraying the calculated heat loss is due to the two different methods used in calculating the infiltration loss. As has already been pointed out, actual pressure drops across the window show the 21st floor to be at about the neutral zone, while THE GUIDE method assumes the neutral zone to be at mid-height or 29 ft higher than the 21st floor. This 29 ft, plus the 15 mile normal wind used in THE GUIDE method of determining the infiltration loss, is responsible for the difference in the two curves.

The curves in Fig. 7 showing the heat supply and the calculated heat loss for the 32nd floor agree remarkably well with the exception of the 13.3 F test point. There is no known reason for this discrepancy.

Fig. 8 shows the amount of heat which would be required to maintain the test rooms on the various floors at 72.6 F if the exposed wall and glass areas were the same as the 8th floor test room. It is evident from Fig. 8 that it requires more heat to maintain a temperature of 72.6 F in the 8th floor test room than it would to maintain this same temperature in a room on the 21st or 32nd floor having the same exposure.

Fig. 9 shows the actual pressure difference across the windows for the three test rooms plotted against the outside temperature with the wind velocity and direction indicated for each test point. For the purpose of comparison, the theoretical pressure difference due to temperature difference for each test room is also plotted. In obtaining the values for the theoretical curve, the effective head was measured from the mid-height of the building. The theoretical pressure difference does not include any wind effect and represents the maximum pressure difference that could exist with zero flow.

CONCLUSIONS

It is evident from the curves of heat supply in Fig. 8 that it requires more heat to maintain the same room temperature on the lower floors of this building, for a given exposure, than it does on the upper floors. The difference in the amount of heat necessary to heat the 32nd floor test room is approximately 40 per cent less than that required to heat the 8th floor room, when expressed on the same amount of exposed wall and glass.

THE GUIDE 1934 method of calculating the heat loss for the 8th floor, which is below the neutral zone, does not give values of heat quantities that compared with those obtained by these tests. The portion of this difference that is due to infiltration cannot be definitely stated at this time, but it is hoped that the future tests being planned will answer this question.

No. 994

WIND VELOCITIES NEAR A BUILDING AND THEIR EFFECT ON HEAT LOSS

By F. C. Houghten,* J. L. Blackshaw ** *(MEMBERS)*, and Carl Gutberlet †
(NON-MEMBER)
Pittsburgh, Pa.

THE estimation of the heating requirements of buildings is the most fundamental practice of the heating and ventilating engineer. It has been the subject of continuous study since the heating of buildings became an important branch of the engineering industry. The determination of the heating load of a building requires *first,* knowledge concerning heat transfer and infiltration through the various types of construction involved for a practical range of temperature differences and wind velocities; *second,* knowledge concerning the weather conditions to which the building will be subjected; *third,* knowledge concerning the proper application of the items listed in formulae to give the resultant heat loss from the completed building or room under consideration.

During recent years research has been directed more intensively toward the extension of the available data on heat transfer and infiltration, and as a result, these data are now better understood than the facts concerning the probable temperatures, wind velocities, sun effect, heat capacity, and other environmental conditions and their use in the calculation of the final results. This subject was approved as a major research project by the Committee on Research last summer, and a Technical Advisory Committee under the chairmanship of D. S. Boyden was appointed.

A great deal of consideration has since been given to plans for this study by the Committee, including the collection of data on the over-all heat demands of existing buildings, an analysis of weather bureau data for various localities with a view of determining the combined effect of wind and outside temperature, and more intensive studies of the heat loss from a few individual rooms in existing buildings. It was agreed to concentrate for the present on the third type of study, namely, investigations of individual rooms in existing buildings. Data were collected during the past heating season in buildings in a number of

* Director, Research Laboratory, American Society of Heating and Ventilating Engineers.
** Research Engr., Research Laboratory, American Society of Heating and Ventilating Engineers.
† Research Assistant, Research Laboratory, American Society of Heating and Ventilating Engineers.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buck Hill Falls, Pa., June, 1934, by F. C. Houghten.

geographical locations, including the study here reported, which was made in eight rooms in the Grant Building, Figs. 1 and 2.

The Grant Building is a modern 40-story office building of the set-back skyscraper type built in 1926. The building is normally heated by a differential vapor system using enclosed copper convectors below all windows. Control

FIG. 1. THE GRANT BUILDING, SHOWING LOCATIONS OF SIX OF THE EIGHT ROOMS USED IN THE STUDY

of heat supply is obtained by varying the pressure of the steam supplied to the system from a small positive pressure in cold weather to the required vacuum in mild weather.

The rooms were chosen so as to give data on the heating demand of the different sides of the building on a given floor as affected by outside weather conditions and data concerning the relation of elevation to heat demand due to exposure to wind and the chimney effect of the building. Three rooms were

chosen on the 7th floor; 726 facing Third Avenue or the southwest, 707 facing Grant Street or the northwest, and 705 facing Fourth Avenue or the northeast. No typical rooms were available for test on the fourth side of the building which was devoted largely to corridors and elevator space. The three rooms on the 7th floor were chosen to represent different exposures in the lower portion of the building, where the heating demands might be affected by other buildings in the vicinity.

Buildings across the streets which may affect wind movement near these rooms and their heights are as follows: Across Third Avenue and opposite room 726 buildings extend to the sixth floor of the Grant Building; across the corner of Third Avenue and Grant Street there is a three-story building; the lot across Grant Street near Third Avenue is vacant; across Grant Street and opposite room 707, across the corner of Grant Street and Fourth Avenue, and across Fourth Avenue and opposite room 705, buildings rise, respectively, to the levels of the 9th, 18th and 14th floors of the Grant Building. Third Avenue, Grant Street and Fouth Avenue, are respectively, 37, 88 and 58 ft wide from building to building.

A single room, 1818, was chosen on the 18th floor facing Third Avenue or the southwest, to represent conditions in the middle or the neutral zone of the building.

Four rooms were chosen on the 27th floor; 2718 facing Third Avenue or southwest, 2715 facing Grant Street or northwest, 2703 facing Fourth Avenue or northeast, and 2706 on the north corner facing both Grant Street and Fourth Avenue. These rooms were chosen to represent the different directions of exposure in the upper portion of the building. With the exception of the corner room, 2706, all rooms had but one exposure, with similarly heated offices on either side as well as above and below. The three rooms facing Third Avenue on the 7th, 18th and 27th floors were similarly located with respect to the plan of the building so as to eliminate the effect of slight variations in location on the floor.

Each room studied was equipped with electrical heaters located directly below the copper steam convector with the cabinet extended downward to 3 in. from the floor so as to enclose the electrical heater. The convection currents of air passed upwards through the steam heating unit and into the room through the same grille which is used when heating the room with steam. The arrangement of the electrical heaters with respect to the convector and its enclosure, the window and other details is shown in Fig. 3, and was observed by the use of smoke to give convection currents of air into the room of the same temperature, velocity and direction as given by normal operation of the steam system.

The electrical heaters were arranged so that a capacity of 2,560 Btu per hour was always controlled by an "on" and "off" thermostat located 36 in. above the floor in the center of the room and shielded from view of the window and convector cabinet. A simple arrangement of switching made it possible to add additional or auxiliary heating capacity in units of 640, 1,280, and 2,560 Btu per hour, according to the heating demand. This gave a very satisfactory and flexible control without the disturbing effect of throwing the entire heating load of the room on and off. Each time upon passing through the rooms

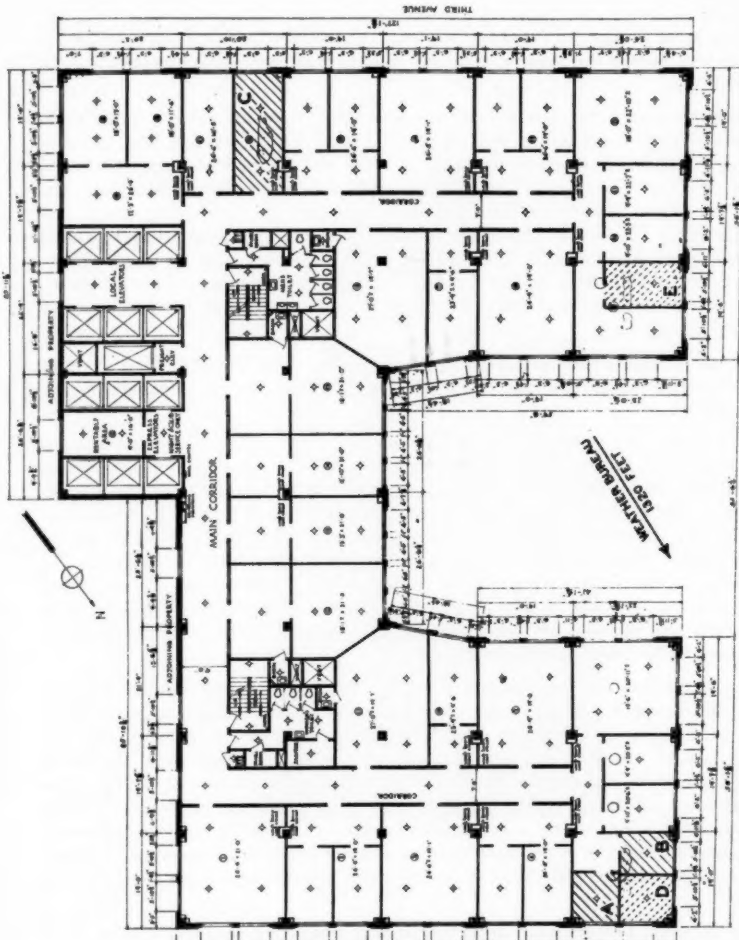


FIG. 2. PLAN OF FOURTH TO TWENTY-THIRD FLOORS OF GRANT BUILDING SHOWING ROOMS *A*, 705; *B*, 707; *C*, 726 AND 1826. ROOMS 2703, 2706, 2715, AND 2718 ARE SIMILARLY SITUATED TO *A*, *D*, *E*, AND *C*, RESPECTIVELY, BUT SET BACK FROM THE BUILDING

the attendant adjusted the auxiliary heating capacity in an attempt to make it and that controlled by the thermostat just a little greater than sufficient to maintain the desired room temperature. All current supplied to each room for the heating units, light and other uses passed through an integrating watt-hour meter having dials which could be read to $\pm$ 5 watt-hours or 17 Btu.

Temperatures were observed by thermocouples, at various heights between the ceiling and floor in the center of the rooms, at a single location in each of

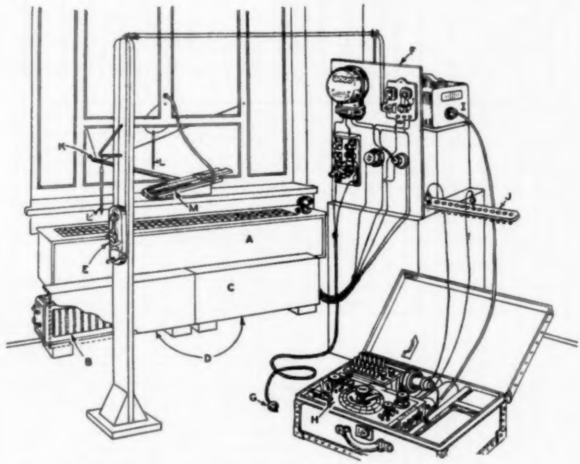


FIG. 3. TEST ROOM SHOWING HEATER AND TEST EQUIPMENT

A—Original convector cabinet
B—Electric heater
C—Extended convector cabinet
D—Air inlet to cabinet
E—Thermostat at 36-in. level
F—Control panel
G—Power line
H—Precision potentiometer and auxiliary equipment arranged for portable use
I—Galvanometer
J—Selective thermocouple switch
K—Thermocouple for observing air temperature 3 ft from outside of window
L—Thermometers for observing inside and outside air temperatures
M—Inclined manometer for observing pressure difference across window

the adjoining rooms, of the surface of the ceiling and floor in the middle of the test room, the inside surface of each of the four walls of the test room, the glass surface, the outside air 3 ft away from the window, and of the outside wall surface of the building. Some of these temperatures were checked by calibrated mercury thermometers. The pressure drop through the window was observed by an inclined manometer.

Wind velocities three feet away from the building wall were observed by an anemometer held out of the window of a room adjoining each of the test

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W
		Weather Bureau Data													Data From Room 2703							
		Wind				Temperature Fahrenheit						Heat Flow Btu/Sq Ft/Hour Through Glass								Percentage		
			Velocity-MPH																			
Test Number	Date – 1934	Direction	Average	Maximum	Minimum	Average	Maximum	Minimum During Test Period	Min. From 10:00 PM To Following 10:00 PM	* Precipitation ~ Inches	Solar Energy on Horizontal Surface ~ Btu/Sq Ft/Day	Maximum	Time of Max.	Av. Condition	Velocity 3 Ft From Window at Time N ~ MPH	Outside Air Temperature at Time N ~ °F	Glass Heat Flow ~ Btu/Sq Ft/Hour/°F ~ Based on P,Q,+70°F	f_o ~ Btu/Sq Ft/Hour/°F	Glass Heat Flow ~ Btu/Sq Ft/Hour/°F ~ Based on Q,S,+70°F	$\frac{O}{M}\times100$	$\frac{R}{M}\times100$	$\frac{T}{M}\times100$
1	Jan 18	SE	9	12	6	30	38	19	19			58.5	9:00A	46.0								
2	19	W	12	18	4	37	40	32	32	T		46.5	9:00A	39.6								
3	20	W	8	12	2	40	46	28	28		638	44.0	7:30A	33.8								
4	22	SE	16	22	13	47	52	41	40	.01	420	35.0	7:00A	29.3								
5	23	W	22	29	14	38	46	30	30	.01	606	49.3	1:30P	42.3								
6	24	S	8	10	3	40	47	26	26		636	44.3	9:30A	33.8								
7	25	W	16	22	11	42	51	33	33	.01	76	44.3	9:30P	35.5								
8	26	SW	8	12	5.	36	41	29	29		674	45.0	7:30A	38.0								
9	27	SW	12	17	8	43	46	37	35	T	153	39.4	8:30A	32.5								
10	29	W	19	22	15	3	5	1	1	T	802	89.8	9:00A	86.5								
11	30	W	14	18	9	6	9	-1	-1	T	689	88.2	6:30A	79.0								
12	31	SW	15	20	10	28	35	12	9		716	69.8	6:00A	52.2								
13	Feb 1	SW	12	15	9	31	34	2	23	T	77	58.0	10:00P	46.8	6.0	23.5	49.8	3.16	48.2	80.7	85.9	83.1
14	2	N	9	11	5	19	22	15	15	T	215	65.0	6:30A	58.2	11.0	16.5	63.0			89.5	96.9	
15	3	S	7	11	4	20	27	8	8		393	69.2	6:30A	54.2	6.5	9.0	65.5	1.46	47.5	78.3	91.7	68.6
16	5	NE	10	12	5	27	29	21	21	T	587	57.2	10:30P	50.0	6.0	21.4	52.0	2.21	44.2	77.4	90.9	70.8
17	6	W	9	12	5	21	26	12	12		511	62.0	6:30A	56.2	6.0	14.0	59.5	1.60	45.5	90.6	96.0	73.4
18	7	NW	13	15	11	26	28	20	20	T	471	63.0	10:30P	53.5	5.0	19.8	52.0	2.46	49.2	84.9	82.5	78.1
19	8	N	13	15	10	6	12	-4	-4		770	89.5	10:00P	78.2	0.4	-1.6	61.0	2.47	70.0	87.4	68.2	78.2
20	9	NE	10	12	7	2	9	-11	-11		747	97.6	6:30A	79.2	7.8	-11.0	90.0	1.79	69.0	81.1	92.2	70.7
21	10	SE	8	12	4	24	34	1	-1		715	72.0	7:00A	51.5	2.0	1.0	63.8	1.73	59.0	71.5	88.6	81.9
22	12	SW	13	19	9	31	33	30	30	.05	153	50.0	10:30P	47.5	15.0	32.6	46.3	2.95	39.6	95.0	92.6	79.2
23	13	NW	17	22	8	14	20	10	10	T	629	72.3	2:00P	71.0	11.2	14.0	66.2	3.26	61.5	98.2	91.6	85.1
24	14	S	8	11	4	24	32	7	5		481	64.5	7:30A	51.6	3.0	8.6	59.3	1.92	55.0	80.0	91.9	85.3
25	15	NW	14	17	6	36	43	21	21		667	58.8	10:00P	42.0	1.4	21.0	43.5	1.65	40.5	71.4	74.0	68.9
26	17	SE	5	8	4	28	37	12	12		479	66.0	8:00A	43.7	1.0	15.0	48.0	2.07	50.4	72.8	80.0	84.0
27	19	NW	13	18	10	18	32	7	7	.08	200	75.0	10:00P	63.3	5.1	6.8	65.5	2.52	63.0	84.4	87.3	84.0
28	20	W	13	18	6	12	20	-2	-2		913	85.2	8:00A	70.5	3.0	0.3	67.2	1.88	60.5	82.7	78.9	71.0
29	21	SW	8	12	2	34	42	14	13		903	56.0	6:00A	40.2	0.6	15.5	46.5	1.99	49.0	71.8	83.0	87.5
30	23	W	19	26	12	14	18	9	9	T	640	76.6	3:30P	72.2	10.0	13.5	65.3	2.66	57.0	94.2	85.2	74.4
31	24	W	10	13	5	11	15	3	3	T	730	80.5	8:00A	68.5	2.8	8.2	59.0	1.90	54.0	85.1	73.3	67.1
32	26	NW	17	21	10	14	17	7	7	.06	394	72.2	4:30P	71.2	5.0	16.0	56.0	3.10	58.0	98.6	77.6	70.3
33	27	W	11	12	4	6	12	-3	-3		1015	86.4	8:00A	76.0	3.8	2.0	66.3	2.70	70.0	88.0	76.4	81.0
34	28	S	7	12	3	20	28	-1	-1		339	71.2	8:00A	54.3	1.8	3.5	60.4	2.17	61.5	76.3	84.8	86.4
35	Mar. 1	S	7	11	3	34	39	17	17		571	51.0	7:30A	39.5	1.0	16.8	46.5	2.07	49.0	77.5	91.2	96.1
36	2	SE	10	16	5	45	51	29	29	.02	214	49.8	6:30A	29.1	1.0	28.8	36.0	1.35	30.5	58.4	72.3	61.2
37	5	SW	16	30	7	48	56	39	39	.24	711	45.6	10:30P	27.8	7.0	38.0	35.0	1.34	23.6	51.0	66.8	41.8
38	6	SW	21	30	12	35	39	29	29	.02	852	55.8	2:30P	45.7	14.0	31.0	47.8	1.76	33.0	71.9	75.7	59.1
39	7	W	9	11	8	35	37	31	31	T	441	46.2	10:30P	40.0	1.4	34.4	31.8	1.43	27.0	86.6	68.8	58.4
40	8	W	11	18	5	28	30	23	23	.04	203	56.5	6:30P	49.5	4.5	29.2	42.0	2.41	43.0	87.6	74.3	76.1
41	9	NW	14	18	8	26	29	20	20	T	1010	63.5	6:30A	59.3	5.0	21.0	51.0	2.93	51.8	85.5	80.3	81.6
42	10	N	14	18	9	22	28	15	15	.37	302	67.3	10:30P	59.2	9.0	19.5	57.2	1.97	45.0	88.0	85.0	66.9
43	12	S	13	17	9	28	33	14	14	T	1071	69.8	6:30A	51.3	19.0	13.8	72.2	3.56	63.5	73.5	103.4	91.0
44	13	SW	20	34	9	52	58	42	33		1175	36.0	6:30A	23.5	19.0	42.0	36.0	2.32	27.0	65.3	100.0	75.0
45	14	NW	15	18	7	28	34	23	23	.04	652	56.5	10:30A	52.2	10.0	28.0	49.0	2.90	44.2	92.4	86.7	78.2
46	15	SW	8	9	3	36	42	18	18		1005	62.0	6:30A	38.0	3.8	18.5	51.0			63.1	98.0	
47	16	W	14	18	2	49	54	40	39		1231	38.2	6:30A	25.9	2.0	40.8	27.2	1.70	24.5	57.8	61.2	54.1
48	17	SW	17	23	9	62	70	43	38		1229	30.6	7:30A	10.2	2.0	43.0	25.0	1.96	24.3	33.3	81.7	79.4
49	24	NW	8	12	5	32	37	25	25		316	50.4	9:00A	42.5	4.6	28.5	42.5	2.23	39.5	79.3	84.3	78.4
50	26	SE	11	19	5	48	59	37	33	.17	230	39.2	6:30A	24.0	1.0	37.4	28.4	2.41	31.6	56.3	72.5	80.6
51	27	NW	14	17	3	37	56	28	28	.26	169	52.8	8:00P	40.7	4.2	28.0	43.0	1.63	34.6	77.1	81.4	65.5
52	28	NW	9	10	4	34	42	22	22		1356	54.8	6:00A	41.3	3.5	26.0	48.3	3.21	48.0	75.4	77.0	87.6
53	29	N	4	7	2	50	61	29	29		446	42.7	6:00A	20.0	1.2	34.0	31.8	1.95	32.3	46.8	74.5	75.6
54	31	E	11	17	7	47	51	41	41	.11	385	34.2	8:00A	27.5	9.7	41.0	33.4	2.31	27.8	80.4	89.1	71.3
55	Apr. 5	E	12	18	8	58	68	45	45		1508	29.8	7:30A	14.5	11.0	46.0	28.2	2.75	24.8	48.7	94.6	83.2
Av. All Tests			12	17	7	30	36	20	19		608	58.6		47.4	5.7	20.8	50.3	2.25	45.8	74.3	83.6	75.4

* T = Trace of Precipitation

TABLE 1. TEST DATA AND RESULTS

rooms. At a distance of 3 ft from the window the component of the wind velocity perpendicular to the building wall and the horizontal and vertical components parallel to the building wall were observed. A cup anemometer was located on the top of the Grant Building and connected to an electrical counter on the 27th floor. Complete Government Weather Bureau data were available from the Pittsburgh station located 1320 ft away in the direction indicated in Fig. 2.

In making a test, the observer entered each room at about 8:00 a. m., turned off the steam heat which had been left on the night before and turned on the electric heating unit controlled by the thermostat and a sufficient number of auxiliary heaters so that the total was a little greater than the heat required to maintain the desired temperature when the thermostat was on. Observation of the room temperature, the temperatures of the surrounding rooms and the watt-hour meter readings were also made on this tour of the eight rooms. Frequent rounds were made of all the test rooms throughout the day, when these observations were repeated and the necessary adjustments to the temperature of the adjoining rooms or to the thermostats in the test rooms were made.

This procedure was continued until 10:00 p. m., when on the last round the observer turned off all electric connections and turned on the steam radiators before leaving for the night, thus allowing the test rooms to float through the night with the same steam heat supply and control as was maintained in the remainder of the building.

Less frequently, additional rounds were made of the test rooms when temperatures given by the thermocouples and thermometers in and about the rooms were read and the wind velocities, 3 ft outside the window, and the pressure drops through the window were observed.

TEST RESULTS AND OBSERVATIONS

Because of the great volume of data collected between January 18, when the tests were started, and April 12, when the last test was made, and because it was impossible to analyze any considerable number of the results while collecting the data, this report will be limited to an analysis and discussion of the results obtained from the weather bureau records, the wind velocities and temperatures observed 3 ft outside of the individual test rooms, and the temperatures of the glass surfaces. Table 1 gives wind velocities and temperatures obtained from the weather bureau records and observed near the Grant Building and results calculated therefrom for the days studied. It will be observed that the minimum daily temperatures for the period of the tests on these days range from −11 to 45 F. The maximum daily wind velocity ranged from 8 to 34 mph. It should be noted that the minimum temperature and maximum wind did not necessarily prevail at the same time.

Column L gives the total solar energy per square foot of horizontal surface for the period from sunrise to sunset as recorded by the weather bureau in Pittsburgh. Column M gives the calculated maximum rate of heat flow through a window from an inside temperature of 70 F and the most severe concurring outside temperature and wind velocity recorded by the weather bureau for the test periods. These calculations were based upon an inside glass surface film conductance coefficient, $f_i = 1.65$, and an outside coefficient, $f_o = 1.65 +$

0.232 V, where V is the wind velocity. This relation between the outside surface coefficient and the wind velocity is in agreement with values determined by the Laboratory in Pittsburgh [1] and by Professor Rowley [2] at the University of Minnesota. Column N gives the time when the maximum calculated rate of heat flow occurred, which was most often in the early forenoon.

Column O gives the average rate of heat flow through the glass based upon the above considerations and the average temperature and wind velocity recorded by the weather bureau for the period of the tests.

Column P gives the wind velocity and Column Q the temperature observed 3 ft outside of the window for room 2703 at the time of the maximum calculated rate of heat flow. Column R gives the calculated rates of heat loss from a 70 F room to the outside air for the observed wind velocities and temperatures given in columns P and Q.

The rate of heat flow, H, through a window may be calculated from the difference between the inside air temperature, t_3, and the glass surface temperature, t_2, if the inside surface film conductance coefficient f_i is known, by the formula:

$$H = f_i\ (t_3 - t_2) \tag{1}$$

Likewise, the outside film conductance coefficient f_o is given by the formula:

$$f_o = \frac{H}{(t_2 - t_1)} \tag{2}$$

where t_1 = the outside temperature

or

$$f_o = \frac{f_i\ (t_3 - t_2)}{(t_2 - t_1)} \tag{3}$$

Column S gives the values for the outside film conductance coefficient based upon the inside room air temperature and the glass surface temperature observed at the time of the maximum calculated rate of heat flow given in column M. Column T gives the rate of heat flow through the glass based on the values of f_o given in column S, a value of $f_i = 1.65$, an inside air temperature of 70 F, and the observed outside temperature 3 ft from the window.

Columns U, V and W give the percentages which the rates of heat flow given in columns O, R and T, respectively, are of the maximum rate of heat flow given in column M.

These percentages show that the heat flow through the window from a 70 F room calculated from the observed temperatures and wind velocities outside of the window is considerably lower than that calculated from the weather bureau data. When the rate of heat flow is calculated from the observed relation between the inside air temperature in the center of the room and the glass surface temperature and the accepted inside film conductance coefficient, the discrepancy between this value and that calculated from the weather bureau is

[1] Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott. A.S.H.V.E. TRANSACTIONS, Vol. 37, p. 301.

[2] Surface Conductance as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. A.S.H.V.E. TRANSACTIONS, Vol. 36, p. 429.

increased. The average for all tests shows the heat flow calculated from the observed temperatures and wind velocities outside of the window to be 83.6 per cent and the heat flow calculated from the inside film conductance coefficient to be 75.4 per cent of that based upon weather bureau observations. While these facts cannot be accepted as final proof of error in the practice of basing heat loss calculations directly on weather bureau data, they do represent strong evidence of such an error.

Considerable variation in the magnitude and direction of the wind velocity observed 3 ft outside the window was apparent. It was also apparent that these velocities were usually lower than those recorded by the weather bureau. Fig. 4 shows the relation between the weather bureau wind velocities and those observed 3 ft outside of rooms 707, 2706, 2718 and 2715, for all observations

Wind Dir. / Room	North				Northwest				West				Southwest			
	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%
705	23.5	39.0	8	0	27.1	60.0	20	1	35.4	88.0	12	3	36.5	95.0	10	2
707	39.9	79.0	9	3	70.2	122.6	17	11	70.6	105.0	6	4	68.4	117.0	12	9
1826	47.0	75.0	9	3	44.3	81.0	18	7	65.0	105.0	7	3	54.6	102.0	16	9
2718	17.0	25.0	7	0	28.5	62.5	19	1	52.2	98.0	10	3	36.5	100.0	15	2
2715	61.6	91.0	6	3	55.0	100.0	17	10	35.5	71.0	13	2	33.2	66.0	16	2
2703	32.7	50.0	7	0	38.9	73.0	22	4	25.9	52.0	10	1	56.4	106.0	17	9
2706	34.3	44.0	3	0	32.6	60.0	16	2	65.6	100.0	10	7	57.7	120.0	18	9
Dir. / Room	South				Southeast				East				Northeast			
705	38.4	62.0	8	1	70.3	91.0	6	5	90.0	90.0	1	1	90.2	114.0	5	4
707	38.5	98.0	11	3									46.0	55.0	3	1
1826	38.2	59.0	6	1	63.5	100.0	8	5	43.0	58.0	2	1	44.5	68.0	7	3
2718	50.1	77.0	7	2	55.0	92.0	9	5	41.0	50.0	3	1	27.4	39.0	7	0
2715	24.1	84.0	8	1	31.4	85.0	10	2	42.8	59.0	4	1	28.4	54.0	7	1
2703	31.7	94.0	9	2	20.6	52.0	10	1	57.0	75.0	6	4	53.0	68.0	5	3
2706	26.4	54.0	9	1	16.1	40.0	10	0	55.0	78.0	3	1	24.0	26.0	2	0

TABLE 2. WIND VELOCITY 3 FT FROM WINDOWS IN PER CENT OF WEATHER BUREAU VELOCITY FOR VARIOUS WIND DIRECTIONS

in all tests when the weather bureau recorded a northwest wind. The values are given by the points of the arrows which indicate the direction in which the wind outside the window was blowing when observed by a person facing out of that particular window. For convenience in comparison, the lines, which represent respectively conditions where the wind velocity outside of the window was 100 per cent and 50 per cent of that recorded by the weather bureau, are drawn. This relationship is better expressed for the eight directions of wind recorded by the weather bureau and for the different windows listed, in Table 2, which gives the average and maximum percentages of all observed velocities outside of each of the seven windows in terms of the concurrent velocity recorded by the weather bureau. This table does not distinguish between direction of the velocities observed outside of the windows. The total number of observations on which the average is based is also given together with the number of observations for any particular window and wind velocity which exceeded 50 per cent of that recorded by the weather bureau.

Table 2 indicates a decided reduction in the observed wind velocities 3 ft from a window below that recorded by the weather bureau, and also a rather definite relationship between this reduction for any window and the direction

of the wind. While in all cases the average wind velocity outside of the window is considerably less than that recorded at the weather bureau, certain windows, depending upon the direction of the wind, indicate occasional high velocities. This reduction in observed wind velocity near a building must necessarily be an important factor in reducing the actual heat loss below that

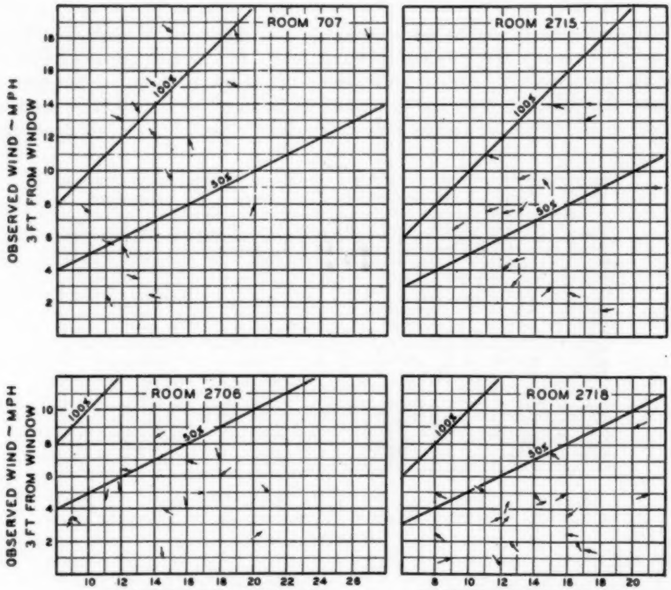


FIG. 4. RELATION BETWEEN OBSERVED WIND 3 FT FROM WINDOW AND WIND RECORDED BY WEATHER BUREAU. ALL DATA FOR A NORTHWEST WIND AT WEATHER BUREAU. ARROW POINTS INDICATE MAGNITUDE AND DIRECTION OF WIND TO OBSERVER FACING WINDOW FROM WITHIN. LINES INDICATE OBSERVED WINDS OF 100 PER CENT AND 50 PER CENT OF WEATHER BUREAU VELOCITIES

estimated from weather bureau data. In this connection it should be pointed out that the velocity a few inches away from the glass or that actually affecting the surface heat transfer may be even a lower percentage of the weather bureau values, because windows are set in several inches from the plane of the outside of the building wall.

The relation between location of a room in a building and the average and maximum winds observed near it as given in Table 2 and Fig. 4 is of special interest. In general, the relation between wind velocity observed near the surface and the weather bureau wind is more or less logical. As an example, when the wind was from the northwest or when it was striking directly into the

broad front of the building it tended to split sideways in each direction, giving high velocities for rooms 707 and 2715 which are near the corners of the building. Most surprising, however, is the fact that the highest velocities were experienced near certain 7th floor rooms where one would have expected shielding due to buildings across the street. As an example, the observed wind velocities were generally highest near room 707, having buildings extending to above this level on the opposite side of the street. Particularly was this true when the weather bureau reported a northwest, west or southwest wind, which seemed to enter Grant Street near Third Avenue and blow in a northeastwardly direction through the canyon formed by the taller buildings on both sides of Grant Street near and beyond Fourth Avenue. These facts do not agree with the frequent practice of assuming a lower wind for sheltered

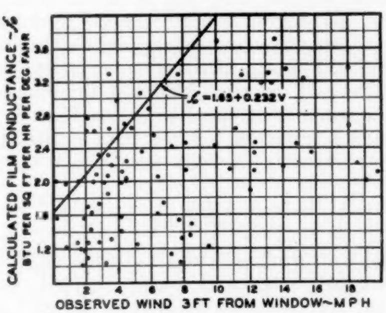


FIG. 5. POINTS SHOWING RELATION BETWEEN CALCULATED f_0 AND OBSERVED WIND VELOCITY 3 FT FROM WINDOW. CURVE GIVING VALUES OF f_0 AS DETERMINED BY FORMULA. DATA FROM ROOM 707

buildings. They seem to lend credence to claims concerning the howling winds through the canyons of New York and other large cities.

Fig. 5 shows the relation between the outside film conductance coefficient, when calculated by formula (3), for all observations on room 707 and the wind velocity observed outside the window. The curve giving values of $f_0 = 1.65 + 0.232\ V$ is drawn for convenience in comparison. It will be observed that most of the values of f_0 calculated from the inside air and glass surface temperatures are considerably lower than those given by the curve for the wind velocity observed 3 ft from the window. Similar data for all other rooms show approximately the same relationship, indicating the probability of a considerably lower actual rate of heat flow through windows than that calculated from wind velocities and temperatures observed 3 ft from the window.

The effect which wind has on the outside film conductance coefficient for a wall or window and therefore its effect on heat transfer is amply demonstrated by the work of the Research Laboratory [1] and Professor Rowley [2] at the University of Minnesota. These laboratory studies, however, gave results for

definitely controlled conditions where the wind was parallel to the wall surface. This does not necessarily apply for natural wind around a building.

The effect which wind has on heat loss from a building differs widely according to variations in wall construction and the percentage of wall and window areas in the building. The effect of wind may be negligible for a well insulated wall, while for a single thickness of glass it may become a very important factor.

The exposed walls of the eight rooms studied in the Grant Building contained 42 per cent of glass and metal sash and frame area, and 58 per cent of masonry walls. The masonry walls had a conductance of 0.413. Assuming an inside film conductance coefficient of 1.65 and an outside brick film conductance coefficient of $f_0 = 2.0 + 0.4\,V$, where V is the wind velocity, the heat transmission coefficient for the wall becomes 0.283 and 0.317 Btu per square foot per hour per degree temperature difference for wind velocities of zero and 15 mph. Assuming glass surface film conductance coefficients $f_1 =$ 1.65 and $f_0 = 1.65 + 0.232\,V$, and allowing no resistance to the flow of heat for the glass itself, the heat transfer coefficient for the glass becomes 0.825, and 1.248 Btu for the same wind velocities. The 15 mph wind increases the heat transfer through the glass to 151 per cent of that for still air, which becomes a considerable percentage increase for the total building.

SUMMARY

This report, dealing with wind velocities and temperatures near the windows of several rooms, and their relation to weather bureau observations and heat loss through windows, represents an analysis of a small phase of the study made in the Grant Building during the past heating season. As such it should be considered as a single step in the analysis of a much more comprehensive study. The results suggest reasons for shortcomings in the present methods of estimating heating requirements of buildings based upon improper application of local weather bureau records to conditions in the immediate environment of a building. The findings here presented should not be considered final and conclusive in demonstrating possible errors in the present methods of estimating heat losses. They should be accepted rather as strong evidence of such errors to be considered in relation to other evidence resulting from more complete analysis of the data already collected in the Grant Building and elsewhere and to future studies, which will probably be made in other types of buildings and in other localities.

ACKNOWLEDGMENT

The authors wish to acknowledge the valuable assistance rendered the Research Laboratory by the Grant Building Management in making the rooms and other facilities available and in otherwise helping to make the study a success. Acknowledgment is also due to Prof. John A. Dent and the Mechanical Engineering Department of the University of Pittsburgh for cooperating in the analysis of the data through the work of student assistants, and for the assistance of Mr. Fred Hogue, a graduate student of the University of Pittsburgh, who aided in the investigation and who is preparing a thesis on the subject for the degree of Master of Science.

DISCUSSION

W. C. RANDALL: I am not surprised to learn that the authors of this paper observed that there were no particular relations between wind velocities recorded at the observatory and those found on the face of the building at various locations.

In our investigations, from which data were received for the paper,[3] we noted that the pressure effect of the wind was greater on the lower floors because it apparently was harder for the wind to spill around the buildings and get away and, therefore, build up pressure. As we measured the pressure drop between the outside and the inside of the building higher up in the building we found, generally speaking, that it decreased, and was negative, particularly in the upper stories. We also found that the stack effect caused, in a great many of the openings on the upper floors, this reversal of pressure drop, and that floors midway between might either have outflow or inflow, according to the location, particularly around the side of the building. These observations are confirmed somewhat by the findings of the paper under discussion.

This brings up the thought that the chapter on Infiltration in THE A. S. H. V. E. GUIDE 1934 might logically be changed so it would not infer that there was a definite infiltration of a window for certain wind velocities, but rather indicate that it was a method of computing heating equipment and possible maximum heating loads. In other words, if the cubic feet of air was changed to a heat factor, the method of making a computation would be more simple and the inference, I believe, would be more correct.

G. D. WINANS: It is interesting to note the results obtained by measuring the wind velocities 3 ft from the building. In the experiments for the paper, Influence of Stack Effect on the Heat Loss in Tall Buildings,[4] by Axel Marin, attempts were made to measure the wind velocities close to the building with no satisfactory results due to the swirling effect of the wind at the building surface.

As explained in the aforementioned paper, measurements were made of the pressure difference across the windows of the various rooms by means of draft gages. To determine the direction of the air movement on the outside of the building, large volumes of tobacco smoke were discharged to the outside of the building through the draft gage connections. The movement of the smoke was a rapid upward one parallel with the building wall in all 3 test rooms. This was done while the wind direction was normal to the building wall. The outside connections of the draft gages were about 4 in. from the face of the wall.

W. W. TIMMIS: This paper demonstrates very clearly the necessity for conducting much more of our testing under actual field conditions than has been the practice heretofore. It is becoming increasingly obvious that the standards developed as a result of our laboratory investigations cannot in many cases be successfully applied under actual working conditions.

When we begin to study the operation of heating systems as systems, and to study the system in relation to the building it is called upon to service, we shall begin to find out many things about heating which have not heretofore been apparent.

The matter of proper distribution of heating elements in the various rooms of a building has assumed much greater importance during the past 5 or 6 years because of the development of control systems in which control is obtained by regulating the flow of heating medium at the source, or of dividing the building into a number of zones and controlling each zone by regulating the flow of heating medium at the entrance to each of the zones into which the piping is divided.

---

[3] Pressure Difference Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 83.
[4] A.S.H.V.E. TRANSACTIONS, Vol. 40, p. 377.

In buildings in which this type of control is utilized it is of the utmost importance that the heating elements be properly proportioned to the spaces they are intended to serve. If they are not properly proportioned, the result is that in order to maintain the spaces which are relatively under-radiated at a desirable temperature, it is necessary to overheat the spaces which are correctly proportioned, and to grossly overheat spaces which are over-radiated.

It is incumbent upon all of us who are working with systems of this type to forward such pertinent field data as may be obtained to the Committee and I for one propose to do so.

No. 995

OPERATING RESULTS OF AN AIR CONDITIONING SYSTEM COMPARED WITH DESIGN FIGURES

By JOHN R. HERTZLER* (*MEMBER*), NEW YORK, N. Y.

PRIOR to 1932 and since 1926, the Union Dime Savings Bank had been satisfactorily ventilated by means of a supply and exhaust ventilation system of the central station type, installed with indirect vento cast iron radiation for winter heating and unit type air filters for year-round cleaning of the air supplied for ventilation purposes. In spite of a large fan capacity and a rapid air change, inside summer conditions were not conducive to human comfort so that an air conditioning system, designed to provide year-round temperature and humidity control was installed, utilizing the existing ventilation system as far as practicable. The installation was made without interfering with the normal banking business in the spring of 1932 and has been in continuous operation since that time. Both the old and the new designs were prepared by Otto E. Goldschmidt, consulting engineer.

It shall be the purpose of this paper to describe the engineering calculations involved in the selection of the refrigeration equipment and to show in general a comparison of the test results and the theoretical analysis of the heat load conditions obtaining.

GENERAL DESCRIPTION OF BUILDING AND OLD EQUIPMENT

Prior to 1926, the bank building, Fig. 1, embraced a property approximately 100 ft square with an inside clear ceiling height of approximately 47½ ft, to which was added at that time, two floors of the adjacent office building, adding approximately 5,800 sq ft of floor area, 22 ft high. In the west side of this addition, a mezzanine level was created to include a director's room and a public room.

The original 10,000 sq ft of floor space with the 47½ ft high ceiling was ventilated by a separate supply and exhaust ventilation system complete with cast-iron radiation and air filters, direct radiation under thermostatic control supplementing the indirect heating for winter temperature control.

The construction of the building included five double glass windows each 25 ft high by 14 ft wide and a glass enclosed entrance vestibule with two

* Air Conditioning Engineer, York Ice Machinery Corp.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buck Hill Falls, Pa., June, 1934.

revolving doors. The 6th Ave. Elevated Railway, east of the building, presented a noise and dirt problem, necessitating equipment and construction as outlined.

The west addition was a remodeled portion of the two lower floors of an adjacent office building and was ventilated by a similar supply and exhaust system.

Several basement rooms, including locker rooms, recreation room and the

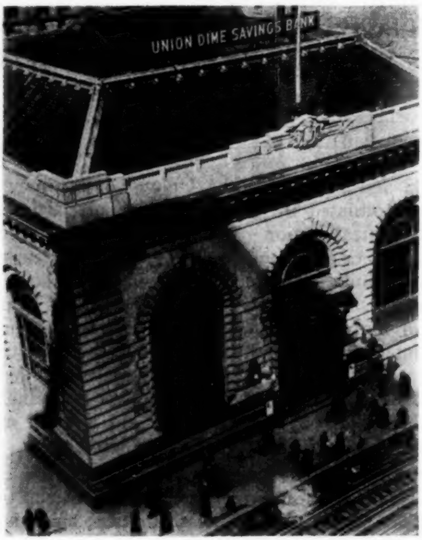


FIG. 1. UNION DIME SAVINGS BANK, NEW YORK, N. Y.

vaults were also supplied by the existing ventilation systems. The toilets were furnished with a separate exhaust system, discharging through the roof.

SELECTION OF AIR CONDITIONING EQUIPMENT

In estimating the heat load imposed on the building in the summer time, the usual calculations in accordance with standard practice, including sun effect on exposed wall area and roof space would be as follows:

INSIDE TEMPERATURE CONDITIONS		OUTSIDE TEMPERATURE CONDITIONS
80 F	Dry bulb temperature.	95 F
67 F	Wet bulb temperature.	75 F
50 per cent	Relative humidity.	38 per cent
60 deg	Dew point.	66 deg
77.3	Grains of moisture per pound.	95.9
13.85	Cubic feet per pound.	14.29

	Sensible Heat Load	Btu per Hour
North Wall	5500 sq ft × 0.25 *U* × (95 deg − 80 deg) =	20,600
East Wall	3630 sq ft × 0.25 *U* × (90 deg − 80 deg) =	13,600
Glass E. Wall	1150 sq ft × 0.45 *U* × (95 deg − 80 deg) =	1,770
South Wall	3650 sq ft × 0.25 *U* × (95 deg − 80 deg) =	13,700
Glass S. Wall	1550 sq ft × 0.45 *U* × (95 deg − 80 deg) =	10,500
West Partition	2176 sq ft × 0.3 *U* × (95 deg − 80 deg) =	9,800
West O. S. Wall	374 sq ft × 0.25 *U* × (95 deg − 80 deg) =	1,400
Ceiling under Office Bldg.	4000 sq ft × 0.3 *U* × (95 deg − 80 deg) =	18,000
Roof Old Bldg	7728 sq ft × 0.26 *U* × (95 deg − 80 deg) =	30,100
Roof Addition	1800 sq ft × 0.28 *U* × (95 deg − 80 deg) =	7,560
Lighting	13600 watts × 3.415 Btu/watt hour =	46,400
Sun effect, S. Wall	3650 sq ft × 0.25 *U* × 25 deg =	22,800
Sun effect, Glass S. Wall	1550 sq ft × 0.45 *U* × 25 deg =	17,420
Sun Effect Roof	728 sq ft × 0.26 *U* × 25 deg =	50,250
Sun Effect Roof, Addition	1800 sq ft × 0.28 *U* × 25 deg =	12,600
People	400 × 225 Btu/hour/person =	90,000
Outside Air	$\dfrac{450{,}000 \text{ cu ft/hour} \times 0.24 \text{ specific Heat} \times (95-80)}{13.85 \text{ cu ft pound}}$ =	117,000
Fan Hp	15 hp × 2545 Btu/horsepower-hour =	38,200
	Total sensible heat load =	527,700

Btu/hour or 44.0 tons refrigeration

	Latent Heat Load	Btu/Hour
People	400 × Btu/hour/person =	70,000
Outside Air	$\dfrac{450{,}000 \text{ cu ft/hour } (95.9 \text{ Gr/pound} - 77.3 \text{ Gr/pound} \times 1060)}{13.85 \text{ cu ft/pound} \times 7000 \text{ Gr/pound}}$ =	91,500
	Total latent heat load =	161,500

Or 13.5 tons refrigeration

Total Refrigeration Requirements

Sensible Heat Load by Calculations	44.0 tons
Latent Heat Load by Calculations	13.5
	57.5 tons
Chilled Water Pump Horsepower & Water Line Losses	5.0 tons
	62.5 tons

Neglecting the basement vault, locker room and recreation room which are relatively unimportant from the standpoint of heat gain, because of the negative effect of the ground, the calculations would show the necessity for installing a refrigerating plant having a capacity of 62.5 tons of refrigeration.

It is to be noticed in Fig. 2 however, that all air supplied to the conditioned space is introduced through outlets not more than 8 ft above the floor line, above which point is the source of considerable of the heat gain. It would be expected that the upper portion of the 47 ft 6 in. high space in the old portion of the building could be stratified and could be considerably higher in temperature than the breathing zone, without causing any detrimental effects.

If the heat gain in the upper two-thirds of the main bank were neglected, the following deductions could be made from the sensible heat load as tabulated:

North Wall	13,800 Btu/hour
East Wall	9,100 Btu/hour
Glass E. Wall	3,885 Btu/hour
South Wall	9,170 Btu/hour
Glass S. Wall	5,250 Btu/hour
Roof Old Bldg	30,100 Btu/hour
Chandelier Lighting	23,200 Btu/hour
Sun Effect—S. Wall	15,300 Btu/hour
Sun Effect—Glass S. Wall	8,710 Btu/hour
Sun Effect—Roof	50,250 Btu/hour
Sun Effect—Roof Addition	12,600 Btu/hour
	181,365 Btu/hour
	= 15.1 Tons refrigeration

These figures indicate that approximately 15 tons of refrigeration could be saved by introducing the air supplied for summer cooling within 10 ft of the floor line. This would be a net load of 47.4 tons of refrigeration, allowing for the same pumping and water line losses as originally suggested.

Type of Equipment Installed

The selection of refrigeration equipment installed was made primarily on the basis of the low operating cost and automatic operation afforded by a freon water cooling system, furnished with a two-speed squirrel cage compressor-motor for automatic operation from the chilled water temperature supplied two air washers.

One air washer was furnished to operate with each of the main supply blowers. The exhaust fans were disconnected and a single supply fan was used to circulate the air in each of the two systems. With the added resistance, the large system was estimated to be capable of delivering 25,000 cfm and the small system—15,000 cfm. The amount of air passed through the air washers was limited to 10,000 cfm in system 1 and 6,700 in system 2.

The automatic summer control furnished, consisted of automatic operation of the spray nozzles in two-thirds of the cross-sectional area of each air washer spray chamber for dry bulb temperature control and control of the spray water temperature by automatic refrigeration capacity regulation for humidity control.

Seasonal change-over thermostats were also provided to automatically throw the control of the chilled water and dehumidifier dewpoint temperature from the refrigeration system to automatic outside air volume dampers which operate to admit sufficient outside air for cooling purposes when the outside air temperature is low enough. In this way, maximum economy of operation is obtained automatically.

Winter temperature control regulating steam supplied the reheater stacks to maintain a 70 F register temperature remained unchanged. The winter humidification is controlled from the dehumidifier dewpoint by regulation of the steam supplied closed water heaters in the chilled water connections at each dehumidifier.

The freon water cooling system as supplied, was 50-tons capacity, when cooling water as required to produce the specified maximum inside temperature and humidity conditions and when supplied with city water for condensing in an amount of 1¼ to 1½ gpm per ton of refrigeration.

The compressor was operated automatically by a two-speed squirrel cage motor 60 hp rating 1200-600 rpm; the machine starting, stopping and changing

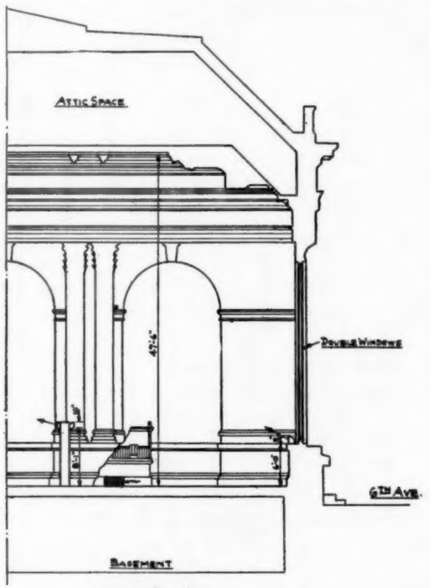


FIG. 2. SECTIONAL VIEW SHOWING AIR OUTLETS

speed to maintain an approximately constant chilled water delivery temperature from the closed shell and tube water cooler. The operating cycle has been adjusted so that capacity changes are automatically made on a 20 to 40 min interval. Under actual test, the efficiency of the compressor-motor at full speed was found to be 87 per cent and at one-half speed, 83 per cent. Since the square feet of evaporator surface and condensing surface per ton of refrigeration is doubled at one-half load, the kilowatt input of electrical energy per ton of refrigerating effect delivered is not increased but decreased for partial load operation of the refrigeration plant. It can be stated that for doubling the amount of condensing surface from 9 sq ft per ton to 18 sq ft per ton, the brake horsepower/ton in a freon system of this kind would be decreased 9 per cent, the evaporating temperature remaining constant. For increasing the evaporator temperature 5 deg from 35 F to 40 F, the brake horsepower/ton

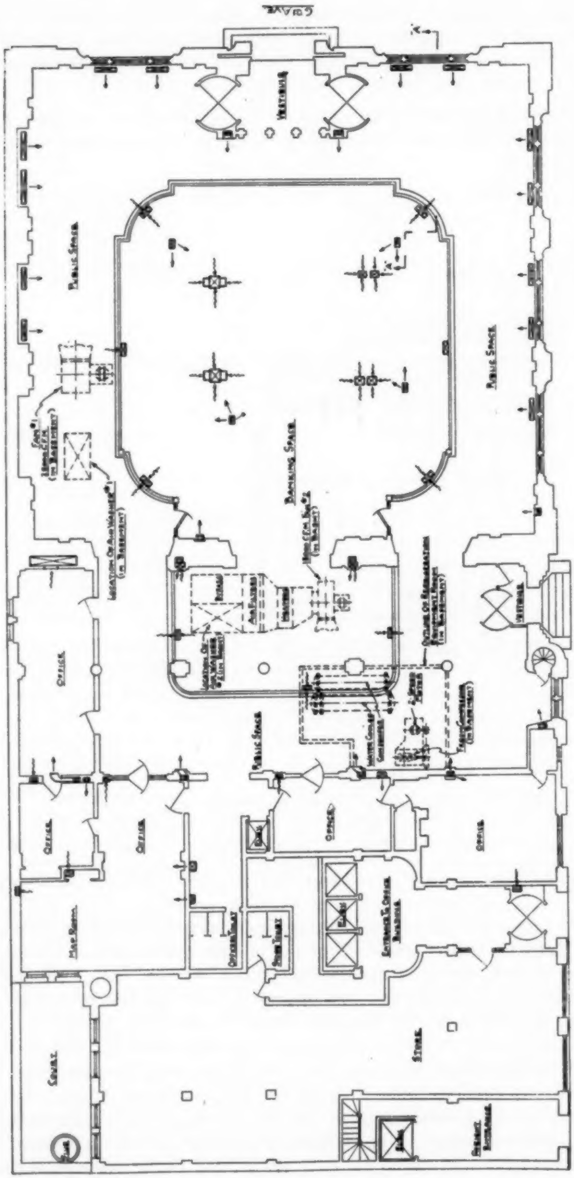


FIG. 3. MAIN FLOOR PLAN WITH AIR CONDITIONING EQUIPMENT SHOWN IN OUTLINE

would be decreased 9½ per cent, the condensing temperature remaining constant. The combined effect of doubling the condenser and evaporator surfaces per ton of refrigeration would therefore decrease the brake horsepower per ton 17.7 per cent, whereas the compressor-motor efficiency drops only 4 per cent for half speed operation.

In addition to the efficient speed control described, a manually operated capacity reducing bypass valve was furnished for each of the two compressor-cylinders, which permits of further capacity reduction to 25 per cent maximum capacity without an increase in full load kw—input per ton of refrigeration effect delivered. For one-half load at full speed, the compressor-motor efficiency was 82 per cent and for one-half load at one-half speed, the motor efficiency was 76 per cent. It has been found from experience that 25 per cent maximum capacity operation has not been required under the actual conditions of operation on the job, the normal start-stop, full speed, one-half speed control furnishing satisfactory operating cycles, even under operation at the smallest loads.

The condenser furnished was of the multipass horizontal closed shell and tube type, as indicated on the first floor plan, Fig. 3. The evaporator furnished was of the flash type, the liquid refrigerant being sprayed over the water tubes of the shell and tube cooler by means of a vertical liquid refrigerant recirculating pump.

The refrigeration system described was installed under the banking room as indicated in Fig. 3, the entire equipment being enclosed in a separate room, ventilated to the outside. The ceiling of this room was covered with an acoustical material to prevent the possibility of any mechanical noises being transmitted to the office space above.

The air washer arrangement outlined on Fig. 3, shows the air filters in series with the air washer. In the rearrangement of the air filters on systems 1—in order to equalize the air resistance, the filter sections were installed in the bypass or recirculating connections to the fan and in the outside air intake. On this system, the return air passing through the air washer is not filtered.

Air conditioning systems are seldom called upon to operate under design conditions but it so happened that July 1, 1932, at the beginning of a quarter, a busy banking day, provided maximum outside temperature and humidity conditions as can be noted from Table 1.

For measuring the refrigeration effect, plate orifices were installed in both the chilled water circuit and the condensing water circuit for measuring the quantity of water in circulation, and temperatures of the water in and out of the condenser and water cooler were recorded regularly to complete the check.

The figure used for tonnage by condenser water readings was 30 gal degrees per minute per ton or 250 Btu min per ton.

For the 12 noon readings, the refrigeration capacity by chilled water measurement was 36.1 tons; by condensing water measurement 37 tons. At 1 p. m. the refrigeration by chilled water was again 36.1 tons; by condensing water 35.8 tons.

For the 2 p. m. and 4 p. m. readings, the refrigeration load by chilled water

measurement was 43.5 tons; by condenser water measurement 42.4 tons at 2 p. m., 41.3 tons at 4 p. m.

The figures cited check quite well and prove quite definitely that the upper section of such a building can be stratified in summer cooling proposition of this nature.

The quantity of outside air admitted for ventilation was rather definitely established by measurement on the outside air intakes and by the capacity of the toilet exhaust systems and the remainder of the factors considered in the calculations were all rather close to the assumed maximums.

The inside temperature and humidity conditions were below the guaranteed calculated figure of 80 F and 50 per cent relative humidity and during the after-

TABLE 1. JULY 1, 1932

		Time					
		9 A.M.	10 A.M.	12 M.	1 P.M.	2 P.M.	4 P.M.
Outside Temperature Degrees	Dry Bulb	78.4	84.5	90.9	89.5	91.5	91.0
	Wet Bulb	69.4	78.0	75.9	76.5	76.5	78.0
Inside Temperature Degrees Sta. (1)	Dry Bulb	75.8	76.6	77.5	78.3	79.0	77.5
	Wet Bulb	66.0	66.0	67.8	68.0	68.5	65.5
Inside Temperature Degrees Sta. (2)	Dry Bulb	75.6	76.0	77.8	78.0	78.5	77.3
	Wet Bulb	65.8	65.5	67.3	66.9	67.5	65.3
Inside Temperature Degrees Sta. (3)	Dry Bulb	76.3	76.3	78.1	77.3	78.0	78.0
	Wet Bulb	66.0	65.5	67.3	66.8	67.0	66.1
Inside Temperature Degrees Sta. (4)	Dry Bulb	76.0	76.5	77.5	77.8	78.2	78.0
	Wet Bulb	66.0	66.0	67.0	66.6	66.8	66.3
Inside Temperature Degrees Sta. (5)	Dry Bulb	75.8	76.0	77.5	78.2	78.7	78.0
	Wet Bulb	66.3	65.0	67.0	67.0	67.3	66.0
Air Out Wash. Degrees No. 1	Dry Bulb	55.0	56.8	58.6	58.5	57.3	57.5
	Wet Bulb	54.7	56.4	58.0	58.0	57.0	57.0
Air Out Wash. Degrees No. 2	Dry Bulb	52.8	53.7	56.3	56.2	57.2	55.0
	Wet Bulb	52.5	53.5	56.0	55.8	57.0	54.2

No. of people in Bank—300-400 all day.
Sta. (1) North Side Old Bldg.
Sta. (2) South Side Old Bldg.
Sta. (3) North Side New Bldg.
Sta. (4) South Side New Bldg.
Sta. (5) Center of Banking Space.

noon operation, the outside dry bulb temperature ranged from 89.5 F to 91.5 F; for outside wet bulbs from 75.9 F to 78F.

The small difference in simultaneous inside temperature readings shows the effectiveness of the air distribution, the maximum variation in the various readings at five stations ranging from 0.5 F to 1 F on both dry bulb and wet bulb temperatures.

With a sensible heat load in the conditioned space of 22-tons which allows for 150,000 cu ft per hour of infiltration direct to the bank, 40,000 cfm of supply air would be warmed through a range of 6.2 F which further indicates that the supply air at the outlets to the bank would be 73.8 F, while maintaining an 80 F inside temperature. This, of course, is considerably more fan

capacity than is recommended for a new installation, but being available, it has been used during a considerable portion of the operating period. The supply fans are equipped with direct connected direct current fan motors provided with field control for adjustable speed operation, offering considerable operating flexibility. The fans during summer operation are run at a fixed speed determined for velocity air distribution from the supply outlets. In the winter time, much less air movement inside the bank is desirable so that the fans are then operated at lower speeds, at the discretion of the operators.

Compressor Operation Period

The operating cycle of the compressor will be duplicated somewhat as tabulated in Table 2 for an average day.

Table 2. Aug. 30, 1932

High Speed Operation		Low Speed Operation	
Time	Minutes Operation	Time	Minutes Operation
8:05–8:45	40	8:45–9:14	29
9:14–9:32	18	9:32–10:03	31
10:03–10:21	18	10:21–10:52	31
10:52–11:10	18	11:10–11:39	29
11:39–11:59	20	11:59–12:25	26
12:25–12:45	20	12:45–1:11	26
1:11–1:31	20	1:31–1:57	26
1:57–2:17	20	2:17–2:43	26
2:43–3:03	20	3:03–3:08	5
	194 Min		229 Min
	3 hours 14 Min		3 hours 49 Min

Total operation........194 Min @ High Speed
229 Min @ Low Speed

423 Min or 7 hours–3 min
High speed operation—45.9 per cent
Low speed operation—54.1 per cent

This method of operation, automatically accomplished has proven very satisfactory with unusually fine conditions being obtained on operating cycles as indicated.

Operating Costs

Based on a 50 per cent load factor, the power and water costs were predicted within an accuracy of 5 per cent, however, there are several interesting phases of the operating cost of the equipment which can be applied quite frequently in the case of institutions of this type.

First, the fan horsepower after the installation of an air conditioning equipment, was reduced because the old exhaust fans, operated previously in the supply and exhaust ventilation system, were shut down for 12 months of the

year. This power saving over 12 months was found to be more than the cost of the power used by the refrigeration plant for four months of the year.

Second, the installation of air washers with facilities for additional cleansing of the air permitted recirculation of a large portion in the winter time, only sufficient outside air being introduced to balance the toilet exhaust system and to provide adequate ventilation. Because of the smaller amount of outside air introduced in the winter time, a definite steam saving was made, the amount of which more than offset the cost of the city water used for condensing purposes in the refrigeration system.

From the records of the bank the cost of power purchased in 1931 was

FIG. 4. INTERIOR VIEW, WEST ADDITION, LOOKING SOUTH

$6,195.60 and in 1932, only $6,274.16 or an increase of $78.56 for air conditioning.

The cost of steam in 1931 was $2,716.62, and in 1933, the first full 12 months with air conditioning $2,059.28, representing a saving in steam cost of $657.34 for the year or of about 25 per cent, due to recirculation. These figures may vary somewhat from year to year as the records show the cost of steam in 1929 as approximately $2,000.00 and in 1930 as approximately $2,500.00, but the principle of the recirculation will always provide a definite saving.

In 1931, the additional water used for condensing cost approximately $400.00, so that the steam saving and slight power increase still show a net saving after air conditioning of approximately $200.00 per year in operating cost.

No additional help was hired to operate the plant because of its classification as a Class B automatic refrigerating system—having less than 1,000 lb of refrigerant charge.

The air conditioning cost on this installation was then only the interest

depreciation and obsolescence charges, usually approximately 15 per cent of the first cost per annum which is economical, to say the least.

The benefits of winter humidification have reduced absences due to sickness and in general, the improved atmosphere has been conducive to good business and better spirit on the part of both customers and employees the year 'round.

Conclusion

In conclusion, it is to be noted that this air conditioning installation—supplementing an existing supply and exhaust ventilation system—

(1) Was installed to utilize the existing ductwork and supply fans, thus greatly reducing the first cost of the installation and eliminating architectural changes in the banking interior.

(2) Stratified the upper portion of a high ceiling space during summer operation, reducing both the first cost and operating cost of the equipment as required.

(3) While maintaining automatically inside atmospheric conditions, at all times within the zone of comfort, reduced the operating cost below that of a supply and exhaust ventilation system which did not provide summer cooling or winter humidifying furnished by the air conditioning system.

That these principles can be applied to other applications, there can be no doubt, and if full inventory of all such cases is made, it is likely that the ever mounting advantages of air conditioning will become even more evident than is the case today.

DISCUSSION

D. W. McLenegan: I think Mr. Hertzler has presented a valuable idea in describing how the air in the upper part of a high-ceilinged room was allowed to stratify. It would be interesting to know the actual temperature gradient from floor to ceiling which resulted from this plan, and I am wondering whether any temperature measurements along vertical lines were taken with the air conditioning system in operation.

J. R. Hertzler: No actual record of the temperature gradient from floor to ceiling has ever been made, but the temperature in the attic space above the main banking room has not been lowered by the installation of the air conditioning system.

An important question brought up by Mr. Carrier is the effect of building lag on the capacity requirements. The 6th Avenue side of the bank has an eastern exposure and faces a public park so that it is exposed to the sun's rays the entire morning. The heat absorbing capacity of the massive walls is great enough, however, to prevent the internal sensible heat gain from reaching the maximum until the sun has passed around to the south side of the building. This is demonstrated by the measured refrigeration loads quoted in the paper as follows:

	Tons by Condenser Water	By Chilled Water
12:00 Noon	37.0	36.1
1:00 p.m.	35.8	36.1
2:00 p.m.	42.4	43.5
4:00 p.m.	41.3	43.5

The increase in refrigeration load of 18 to 20 per cent from 1:00 p.m. to 2:00 p.m. cannot be accounted for by an increased number of people in the bank or by an increased outside temperature. It is assumed, therefore, that the increase is due to the limit to the heat absorbing capacity of the walls having been reached between 1:00 and 2:00 p.m. without a decrease in wall surface exposed to the sun.

In an installation where there is a single wall exposure the effect of building lag would permit of a reduction in installed capacity but this is not true in the case of the above building.

No. 996

INSULATING VALUE OF BRIGHT METALLIC SURFACES

By F. B. Rowley † (*MEMBER*), Minneapolis, Minn.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers and conducted at the University of Minnesota

THE insulating value of any wall or built-up combination of materials depends upon the thermal properties of its component parts. For the purpose of making a thermal analysis, a wall may usually be divided into sections of homogeneous material, air spaces, and surfaces.

The transfer of heat through homogeneous materials is usually considered to take place by conduction only, although in many cases the transfer is partly by radiation and convection. For instance, in a fibrous insulating material or any material in which there are air spaces, heat is transferred by a combination of conduction, convection, and radiation. In such cases, the structure is usually too complicated to attempt any separation of the total heat transferred into its component paths and the coefficient is considered to be one of conduction only. For exterior surfaces and air spaces, it is often possible and profitable to make a more complete analysis of the heat flow. While such an analysis is often practical, there are many variables and conditions in practice which make the final results only approximate and make check tests desirable.

The purpose of the present discussion is to show by test results the effect of bright metallic surfaces on the heat transmission coefficients, and to point out some of the difficulties in making a theoretical analysis, or in establishing absolute coefficients by test procedure which can be indiscriminately applied to all cases. It is further the object to show reasonable average coefficients which may be used in practice.

Heat is transferred to or from the surface of the material by air contact and by radiation. The amount which is transferred by radiation depends upon several factors one of them being the character of the surface of the material. Until recently, it has seemed sufficiently accurate for practical purposes to consider the materials as having similar radiation characteristics and to use average transmission coefficients where surfaces were involved. Thus such surfaces as paper, wood, plaster, stone, etc., which are used extensively in building construction have emissivity coefficients which vary somewhat and yet which are sufficiently uniform to permit the use of average coefficients inso-

† Director, Experimental Engineering Laboratories, University of Minnesota.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, June, 1934, Buck Hill Falls, Pa.

far as radiation of heat is concerned. The roughness of the surface affects the transfer by convection, but for average conditions this difference has also been neglected in the interests of simplicity in selecting average coefficients. At present, there is an active interest in the use of bright metallic surfaces which have high resistance to the flow of heat by radiation and which give entirely different transmission coefficients than the above mentioned group. In order to meet these conditions, it is necessary to revise the old practice and add new coefficients to the list which will more nearly represent the conditions to be expected when materials are used with low radiating coefficients.

In considering the coefficients for surfaces with high resistance to radiant heat, the same problem exists as for the present common type of building materials. The emissivity varies even for the same material, depending upon the

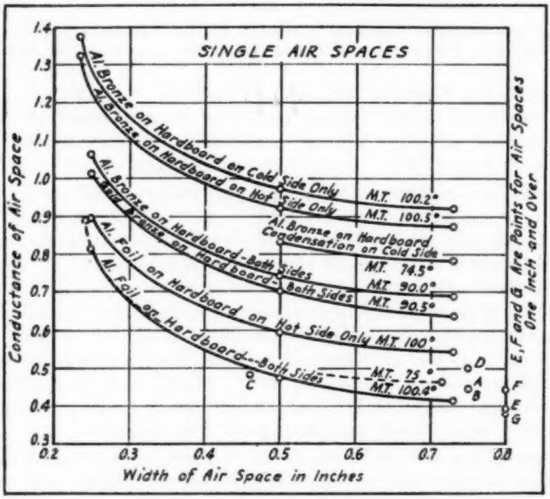


FIG. 1. RESULTS OF INDIVIDUAL TESTS TO DETERMINE AIR SPACE CONDUCTANCE

surface finish. For instance, aluminum foil which has a very low emissivity will not always give the same results under test. The polish and character of the surface is not always the same; in fact, under certain adverse conditions the increased insulating value may be entirely lost. If the surface becomes covered with moisture, dust, or some other foreign material which has a high emissivity coefficient, or if it becomes dull and tarnished, its insulating value will be greatly reduced. Due to the difficulties involved in making theoretical calculations, it seems desirable to derive average coefficients from actual tests which may be expected to hold when the materials are built into a wall. With this purpose in view, a series of test results obtained both by the hot plate method and by the hot box method have been analyzed to determine coefficients which may have practical application.

In making the tests, the standard hot plate and hot box apparatus have been used. The general procedure for determining air space coefficients has been to construct a test section with an air space of the desired thickness and lined on one side or both surfaces with the test material. The conductance of the air space has been calculated by standard formula from the test results which have been obtained for the overall conductance of the built-up section and the conductance of the material exclusive of the air space.

For surface coefficients, the determinations have been by the hot box method only. Since the details of the apparatus, the standard test method, and general

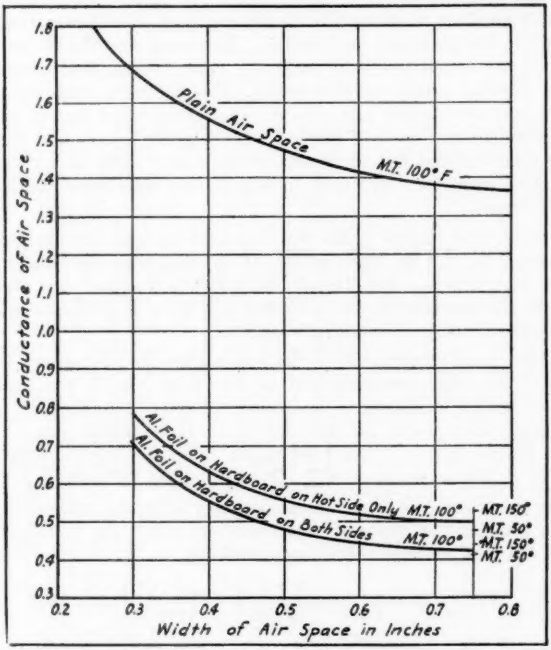


FIG. 2. AVERAGE CURVES FOR 100 DEG MEAN TEMPERATURE

procedure have been reported in previous papers, the results for the present tests will, wherever possible, be reported graphically.

The results of individual tests made to determine air space conductance are shown in Fig. 1. These results were taken from several different sources of test data and include both the hot plate and hot box methods when using different boundary surfaces, as indicated. The most extended single series of tests are represented by the solid-line curves. The dash-line curve represents the second independent series by the hot plate method in which a different stock of aluminum was used as surface lining. These results agree reasonably well with the first series for similar construction, although the conductances are

slightly higher at 0.25 in. and 0.75 in. The test results indicated by points *A, B, C,* and *D* were obtained from tests made at different times by the hot plate method when using aluminum foil selected from independent stocks. The test results as indicated by points *E, F,* and *G* were all obtained by the hot box method. Test conditions and details for the points indicated are given in Table 1.

While the test results of Fig. 1 show a reasonable agreement between the different series of tests, there is some variation which may be attributed to several causes. In the first place, the surface of the material was probably not in exactly the same condition as selected from different stocks. There is also some possible question as to the size of the test area when comparing the results obtained by the hot plate and the hot box. For those results obtained by the hot plate method, the actual test area was 9 in. square which was divided from the guard ring area by a thin strip of wood. For the tests made by the hot box method, the test areas were 3 ft square. It will be noted that in general those results obtained by the hot box method were slightly lower than those obtained by the hot plate method for specimens of the same width of air space and the same boundary material. The area of test section may have had some bearing on this difference.

For all of the test results by the hot box method, with the exception of point *F,* there were 2 in. x 4 in. studs spaced 16 in. on center, dividing the air space through the test section. For Test *F,* the entire 3 ft square test area was open. Undoubtedly, the presence of the studs through the test section would somewhat reduce the effectiveness of the bright surface.

Taking into consideration all of the test values for air spaces lined on one or both surfaces with aluminum foil as shown in Fig. 1, the average curves for 100 deg mean temperature as shown in Fig. 2 have been constructed. For comparative purposes, the 100 deg mean temperature curve for surfaces lined with fibrous material has been taken from the curves published in the A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, pages 165-168.

From the curves on Fig. 2, a comparison may be made between the emissivity coefficients for the two types of materials by using the well-known Stefan-Boltzmann law and making certain assumptions.

$$(1) \qquad H = 0.172(A)\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right](F_A)\ (F_E)$$

In this formula, the angle factor F_A may be taken as 1, and the emissivity factor F_E may be taken as:

TABLE 1. TEST VALUES FOR THE CONDUCTANCE OF AIR SPACES AS USED IN FIG. 1

Test Point Fig. 1	Date of Test	Test Method	Thickness Air Space In.	Surface, Hot Side	Surface, Cold Side	Conductance *G*	Deg. F. Mean Temp.
A	12/6/'33	Hot plate	0.75	Aluminum foil	Aluminum foil	0.449	75
B	6/13/'32	Hot plate	0.75	Aluminum foil	Wood fiber	0.447	90
C	1932	Hot plate	0.46	Aluminum foil	Aluminum foil	0.485	77
D	5/6/'33	Hot plate	0.75	Aluminum foil	Dull black	0.50	75
E	12/6/'33	Hot box	3½	Aluminum foil	Wood fiber board	0.393	40
F	8/4/'32	Hot box	3	Aluminum foil	Aluminum foil	0.44	40
G	10/15/'30	Hot box	1	Aluminum foil	Aluminum foil	0.383	40

TABLE 2. RESULTS OF TESTS BY HOT BOX METHOD TO DETERMINE SURFACE COEFFICIENTS FOR A WALL WHEN COVERED WITH PAPER AND WITH ALUMINUM FOIL

1 Date of Test	2 Wall No.	3 Lining for Both Surfaces	4 Mean Temp. of Test	5 Over-all Co-efficient, U	6 Conductance Surf. to Surf., C.	7 Av. Surface Co-efficient	8 Surface Radiation	9 Surface Convection
							Btu	
10/15/'30	78	Paper	40 F	0.691	3.5	1.72	0.733	0.99 −
10/20/'30	78B	Aluminum foil	40 F	0.46	3.5	1.06	0.068	0.99 +

$$(2) \qquad F_E = \frac{1}{\frac{1}{p_1} + \frac{1}{p_2} - 1}$$

in which p_1 and p_2 are the emissivity coefficients for the materials on the corresponding sides of the air space. If the average emissivity coefficient p_1 for the non-reflecting material, such as wood, paper, etc., be taken as 0.9 and it is assumed that the amount of heat given off by convection is the same for both types of surfaces, the emissivity coefficient for aluminum foil surfaces may be calculated from the test results. This method gives an emissivity coefficient of 0.08 which is slightly larger than that often quoted for aluminum.

The mean temperature variations as shown on Fig. 2 are based on the assumption that the difference in heat transfer for various mean temperatures is entirely due to radiation and not to convection. These are calculated and not test values, but since they are small variations, the error can only be slight. The small difference between the curves for single and double air space linings as shown on Fig. 2 is due to the fact that the foil has such a low emissivity coefficient that a single surface intercepts practically all of the radiant heat, leaving but a small portion as a possible saving for the second surface lining. This is readily shown by an inspection of formula (2) which gives the value of the emissivity coefficient.

Substituting emissivity values in this formula,

$$F_E = \frac{1}{\frac{1}{p_1} + \frac{1}{p_2} - 1}$$

The values for F_E are as follows:

No foil on either surface, $F_E = 0.818$
Foil on one surface, $F_E = 0.079$
Foil on both surfaces, $F_E = 0.042$

Thus, $\frac{0.818 - 0.079}{0.818}$ or 90.3 per cent of the radiant heat is stopped by one surface lining and $\frac{0.818 - 0.042}{0.818}$ or 94.8 per cent is stopped by two surfaces, lined.

Referring to Fig. 1, the difference in the values for air spaces lined on one or both surfaces with aluminum bronze is greater than for those air spaces

lined on one or both surfaces with aluminum foil. This is due to the fact that the emissivity coefficient for bronze is higher than for aluminum and a single surface lining does not intercept so large a percentage of the radiant heat. It will be noted that the transmission is slightly higher for an air space lined on the cold surface than for one lined on the hot surface. This was found in several cases when using aluminum foil, but in most of these cases condensation was shown to be present on the cold surface. Since the emissivity coefficient for water is high, the presence of a water film on the aluminum surface practically destroys its value.

From the curves of Figs. 1 and 2, it is apparent that the variation in conductance for air spaces of different widths is the same regardless of the surface lining, and the air space must be 0.75 in. or more in width to get the full value of heat resistance. This variation in conductance is due to convection as the radiant heat is substantially the same regardless of thickness.

In considering surface coefficients for building material, the usual procedure has been to consider them as unit coefficients without regard to the percentage transferred by radiation or convection. This appears to be sufficiently accurate for average practice with ordinary building materials. However, for those materials with a low emissivity coefficient there may be an appreciable error, especially for low wind velocity. In most cases, these coefficients have been determined and are used for temperature differences between the surface of the wall and the air, which are substantially the same as the temperature differences between the surfaces of the wall and the surfaces of the surrounding objects, and for average practice the radiant heat may be separated from the convected heat by assuming the same temperature drop for each.

Any radiation coefficient which may be selected for test purposes will not apply to all practical applications, due to the many variations in emissivity coefficients both in the particular wall surface under consideration and in the surfaces of surrounding objects. Average values, however, may be taken for those materials most commonly met with and relations obtained which are sufficiently accurate for practical purposes. Thus it seems reasonable to assume 0.9 as the average emissivity coefficient for the materials which surround the outside of a building wall or for those normal materials which are used in inside construction, with the exception of the special cases of bright metallic or highly reflecting surfaces. Under this assumption, the amount of heat radiated to or from exterior surfaces of walls may be calculated by the Stefan-Boltzmann law with a reasonable degree of accuracy. By using equation (1), the factor F_A may be taken as 1, the emissivity of the wall surface material may be selected as an average value for the particular material from which the wall is constructed, and for normal cases the emissivity coefficient of the surrounding objects may be taken as 0.9. On this basis, corrections can be made to existing wall coefficients.

Table 2 shows the results of tests made by the hot box method to determine the differences in surface coefficients for a wall lined with paper which represents the average building material, and one lined on both surfaces with aluminum foil. In order to eliminate as far as practical all heat resistance in the wall excepting that of the surfaces, a wall was constructed of ⅜-in. thick sheetrock, and the only difference in the two walls was that in Wall

TABLE 3. REDUCTION IN AVERAGE SURFACE TRANSMISSION COEFFICIENTS CAUSED BY COVERING THE SURFACE WITH BRIGHT ALUMINUM FOIL

Mean Temperature, Deg. F	Reduction in Coefficient, Btu
20	0.559
40	0.634
60	0.711
80	0.797
100	0.887
120	0.99
140	1.09

No. 78B bright aluminum foil was pasted on to the two surfaces of Wall No. 78.

The surface coefficients in column 7 of Table 2 are the averages for the two surfaces of the wall under test. The radiation coefficients in column 8 were calculated from the Stefan-Boltzmann law by assuming emissivity coefficients of 0.9 for the paper surfaced wall and also for the surrounding objects in both cases, and an emissivity coefficient of 0.08 for the aluminum surfaced wall. The heat transmitted by conduction and convection in each case, as shown in column 9, was obtained by taking the difference between the total heat transferred and that transferred by radiation alone. These tests were previously reported in the A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 491, although the surface coefficients were not analyzed at that time.

The values which have been accepted for surface conductances from building materials may be reduced when using surface finishes which have a high resistance to radiant heat. The reduction will depend upon the emissivity of the surface and upon the mean temperature between the surface and the surrounding objects. It will be independent of air velocity over the surface. For average conditions of aluminum surfaces which are highly polished, it would seem reasonable to make reductions in the transmission coefficients as shown in Table 3 for the different mean temperatures. These reductions, however, cannot be followed blindly, due to the many variables which enter the problem.

In conclusion, it may be said that the generally accepted coefficients for wall surfaces and for air spaces which involve surfaces should be revised if special materials with low emissivity coefficients are to be used for surface linings. The reduction will depend upon the surface characteristics of the material. If the emissivity coefficient is known, the correction can be made with reasonable accuracy by calculation. For practical purposes, the test coefficients may be used for aluminum foil lined with air spaces as given in curves of Figs. 1 and 2, and reductions for surfaces as given in Table 3.

This paper is the result of cooperative research work between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Minnesota.

The author wishes to give credit to Mr. William A. Eckley and Mr. Robert M. Lander for the work which they did in obtaining the test values for those points shown on Table 1 by the hot plate method.

DISCUSSION

J. H. BRACKEN (WRITTEN): Fig. 2 of this paper shows approximate curves for plain air spaces, and for air space with aluminum foil on the hot side and on both sides, at a mean temperature of 100 F. The curve for the plain air space is taken from Fig. 3 of the paper entitled, Thermal Resistance of Air Spaces,[1] by F. B. Rowley and A. B. Algren.

In northern latitudes, a temperature of 100 F exists in the air spaces of the average wall for only a few hours during the year on the sides exposed to the sun and practically never on the other sides. On the other hand, the outside air temperature averages between 25 and 45 deg during the heating season of about 7 months in the northern half of the United States, and the average air space temperature is more nearly 50 deg. Furthermore, in practically all cases a positive temperature of 70 or 75 deg is maintained during the cold weather, whereas a very small percentage of buildings have temperature control in the summer. Therefore, for comparative purposes 50 deg would be a more logical temperature.

If the 50 deg air space curve of the aforementioned paper is superimposed on Fig. 2 of the paper under discussion, the comparative results are quite different. For example, the resistance of a ½ in. air space bounded by ordinary materials at 100 F is about 0.67 whereas at 50 F the resistance is about 0.81 or about 21 per cent greater.

The conductance of the ½ in. air space with foil on both sides at 50 F is about 0.46 (according to Fig. 2 of the paper) and the resistance is about 2.17. The difference in resistance between the ordinary space and the foil lined space therefore is about 1.36, which difference should be fairly constant for this mean temperature for all widths of air spaces if the assumption made in the paper is adopted; namely, that the mean temperature variations as shown in Fig. 2 are based on assumption that the difference in heat transfer for various mean temperatures is due entirely to radiation and not to convection.

On a similar basis, the difference for a mean temperature of 50 deg for one foil surface is about $\frac{1}{0.55} - 0.81$ or 1.01. The 50 deg curves for foil differ only slightly from those for 100 F and consequently the effect of the foil is decidedly less marked than at 100 deg.

Therefore, it appears that the effectiveness of bright foil diminishes as the mean temperature diminishes, which is in accordance with the Stefan-Boltzmann law and furthermore this effect diminishes as the polish and brightness of the foil decreases until the point is reached at which the aluminum foil has no additional value over ordinary surfaces. It should be borne in mind, also, that bright aluminum foil is only effective when exposed to the air. When used between and in contact with other materials it has no insulating value under any conditions.

P. D. CLOSE (WRITTEN): The rate of heat transfer across air spaces apparently depends upon a greater number of significant variables than the rate of heat transfer through solid materials. These variables include (1) mean temperature; (2) temperature difference; (3) position of air space, that is, whether horizontal or vertical; (4) size of air space; (5) width of air space; (6) if horizontal, direction of heat flow, that is, whether up or down; (7) whether the surfaces are smooth or rough and (8) brightness or dullness of surfaces, *i.e.* emissivity.

This paper has supplied much needed data on the theoretical effectiveness of aluminum foil as an insulating medium and should be a valuable guide in future heat transmission calculations where this material is under consideration.

The statement is made in this paper that, "The generally accepted coefficients for wall surfaces and for air spaces which involve surfaces should be revised if special materials with low emissivity coefficients are to be used for surface linings."

---

[1] A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 165.

This paper shows very definitely that clean, highly polished aluminum foil surfaces substantially reduce the rate of heat transfer across air spaces when the surfaces of ordinary building materials are replaced by this material. This also applies to the heat emitted from or received by inside and outside wall surfaces.

The paper very aptly points out, however, that, "Aluminum foil which has a very low emissivity will not always give the same results under test. The polish and character of the surface is not always the same; in fact, under certain adverse conditions the increased insulating value may be entirely lost. If the surface becomes covered with moisture, dust, or some other foreign material which has a high emissivity coefficient, or if it becomes dull and tarnished, its insulating value will be greatly reduced."

On page 418, the following statement appears: "It will be noted that the transmission is slightly higher for an air space lined on the cold surface than for one lined on the hot surface. This was found in several cases when using aluminum foil, but in most of these cases condensation was shown to be present on the cold surface. Since the emissivity coefficient for water is high, the presence of a water film on the aluminum surface practically destroys its value."

It is evident from these statements that the all-important consideration is whether or not the surfaces remain bright or become covered with moisture, dust, etc., or are affected by fumes and smoke resulting from manufacturing processes, combustion of sulphurous fuels, etc.

The atmosphere of metropolitan areas is contaminated by various substances which become attached to building surfaces, both inside and outside. The most obvious example is that of window glass on which impurities in the air accumulate and which must be washed periodically, both inside and out. In regard to air pollution, the following statement on page 207 of THE A. S. H. V. E. GUIDE 1934 may be of interest: "The most conspicuous sources of atmospheric pollution may be arbitrarily classified according to the size of the particles as dusts, fumes, and smoke. Dusts are particles of solid matter varying from 1.0 to 150 microns in size. Fumes include particles resulting from chemical processing, combustion, explosion and distillation, ranging from 0.1 to 1.0 micron in size. Smoke is composed of fine soot or carbon particles, less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health."

Aluminum covered surfaces, whether foil or paint, become coated with tar, carbon, ash and dust and ultimately lose their brightness and lustre. They certainly are not immune to the effects of atmospheric pollution.

Meller and Sisson in an article on air pollution in the March 1934 *Architectural Record* state that, "Scouring may be necessary to rid the surface of soot and tar, but injury to the metal results through abrasion." Referring to chromium, aluminum and corrosion-resisting alloys used on the Empire State, Chrysler and other buildings, they state that, "The griming is as great as with other metals."

It is true, of course, that atmospheric contamination may not affect the surfaces of air spaces within walls to the same extent as inside and outside surfaces, but unless such air spaces are hermetically sealed—and they are not in ordinary building construction—the surfaces will eventually become coated and such air spaces will then have no greater insulating value than air spaces bounded by ordinary materials.

Recently I bought a wrist watch of what I considered to be a good make. For all practical purposes the crystal is air tight and yet a thin film of dirt has formed on the underside of this crystal and on the face of the watch. Certainly the average air space in a wall is not as well sealed as this watch crystal and yet the latter does not preclude the entrance of fine atmospheric impurities.

Manufacturers of aluminum foil make the statement that aluminum is always coated

with a thin, invisible oxide which protects and preserves its reflectivity against corrosion. An invisible oxide film however can hardly be expected to prevent the deposition of dust, smoke, tar, etc.

I am informed that the walls of certain buildings, in which aluminum foil insulation had been applied, were opened to ascertain the condition of the foil after a period of service. The condition of the foil would obviously depend on the tightness of the construction, the location of the building particularly with respect to industrial centers, the length of time the foil had been installed, etc. The majority of the foil installations in buildings have been made in the last year or two.

A few years ago, one of the large manufacturers of household refrigerators adopted aluminum foil insulation. I am informed that the use of this material has now been discontinued. Before definite values are assigned to aluminum foil, it is suggested that this matter be investigated by the Society.

Whenever air spaces of any type are used for low temperature insulating requirements, it is essential that these air spaces be hermetically sealed, otherwise condensation of the water vapor from the warm side will take place on the cold surfaces of the air spaces. Under extreme conditions, the water vapor thus condensed will fill the spaces to a considerable depth.

It is apparent, therefore, that while bright, highly polished aluminum foil surfaces appreciably increase the heat resistance of air spaces and of wall surfaces, the effectiveness of the foil diminishes as the surface becomes coated with foreign substances or corroded by chemical reaction. It is very likely that under practical conditions this coating will ultimately become of sufficient depth to render the bright metallic surface of no additional insulating value over an ordinary building surface. Therefore, for the purposes of heat transmission calculations, it is my opinion that for the present no allowance be made for aluminum foil or other bright metallic surfaces until conclusive information concerning performance in practice is available.

E. R. QUEER (WRITTEN): The insulating value of bright metallic surfaces has attracted considerable attention for the past six or seven years. Much time and energy has been expended in investigating its properties, and good data have appeared in literature.[2-3-4-5-6]

TABLE A

Heat Input Btu/Square Feet Hour	Air Space Boundary: High Side Temp. Deg. Fahr.	Air Space Boundary: Low Side Temp. Deg. Fahr.	Mean Temp. Deg. Fahr.	Temp. Difference Deg. Fahr.	Conductance Btu/Square Feet Deg. Fahr.	Remarks
4.69	46.33	40.37	43.35	5.96	0.785	conventional 3⅝ in. air space
10.10	57.15	46.00	51.50	11.15	0.905	
17.50	71.49	54.46	62.98	17.03	1.028	
4.69	65.70	39.10	52.40	26.60	0.177	1—sheet aluminum foil 2—1.81 in. air spaces
10.10	94.75	44.37	69.50	50.38	0.201	
17.50	134.30	53.20	93.70	81.10	0.216	
10.1	85.2	44.2	69.70	41.00	0.246	aluminum foil torn at top & bottom

[2] Heat Insulation with Aluminum Foil, Professor Dr. Ing. E. Schmidt, *Verein Deutscher Ingenieure* 71, 1395 (Oct. 1, 1927).

[3] Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, L. W. Schad, A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931, p. 285.

[4] Properties of Metal Foil as Insulation, J. L. Gregg, *Refrigerating Engineering*, May 1932.

[5] Surface Absorption of Heat from Solar Radiation, F. G. Hechler and E. R. Queer, *Refrigerating Engineering*, Feb. 1933.

[6] Thermal Insulation with Aluminum Foil, R. B. Mason, *Industrial and Engineering Chemistry*, March 1933.

It is well to keep in mind that even at ordinary temperatures from 55 to 70 per cent of the total heat transfer across air spaces bounded by conventional building materials is by radiation. Adding bright metallic surfaces properly placed will materially reduce the radiant heat transfer across air space, hence they add insulation. To demonstrate the efficacy of a sheet of bright metal foil as insulation, a common construction of an air space was made and the foil inserted to divide the air space into two equal parts. Fig. A shows the experimental assembly. The conductance of the 3⅝ in. air space was measured and found to be 1.1 Btu/hour/square foot/degree Fahrenheit at 70 F mean temperature. The air space was then divided into two equal spaces 1.81 in. thick by inserting a sheet of bright aluminum foil. The conductance measured between the same bounding surfaces as in the previous case

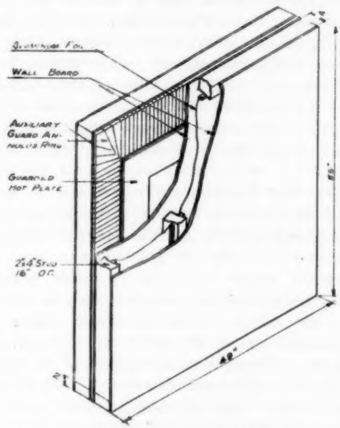


FIG. A. EXPERIMENTAL ASSEMBLY

dropped to 0.202 Btu/hour/square foot/degree Fahrenheit at 70 F mean temperature. The aluminum foil sheet was then torn at the top and bottom of each section, permitting a circulation of air between the air spaces and the conductance increased slightly to 0.246 Btu/hour/square foot/degree Fahrenheit at 70 F mean temperature. These results are presented in more detail in Table A. The conclusion drawn is that by reducing radiation which usually is the major mode of heat transfer at ordinary temperatures, insulation is added to the structure.

A comparison was made to illustrate the thickness of board from insulation necessary to provide the same insulation as that of the aluminum foil in the test. The board form is assumed to divide the 3⅝ in. air space into two equal parts in the same manner as the aluminum foil in the experimental assembly.

Conductance—air space bounded by wall board:

$C = 0.202$ Btu/hour/square foot/degrees Fahrenheit at 70 F mean temperature (by test)
Conductance—thin air spaces—bounded by wall board on one side and board form insulation on the other:
$Ca = 1.1$ Btu/hour/square foot/degrees Fahrenheit
(A. S. H. V. E. GUIDE 1934)
Conductivity of insulation, $k = 0.34$ Btu/hour/square foot/degrees Fahrenheit/inch
Thickness of board form insulation with equivalent conductance to aluminum foil $= x$ inches.

$$C = 0.202 = \frac{1}{\frac{1}{CA} + \frac{1}{CA} + \frac{x}{k}} = \frac{1}{\frac{1}{1.1} + \frac{1}{1.1} + \frac{x}{0.34}}$$

$$x = 1.06 \text{ in.}$$

It is well to caution that this result should be interpreted to mean that one thickness of foil has the equivalent insulating value of 1 in. of board form of insulation, only for the manner in which it was assumed to have been applied. If the board form was applied to the outside of the studs, as it usually is, and compared to the foil applied as in the experimental assembly it would take 1.37 in. of board.

$$C = 0.202 = \frac{1}{\frac{1}{CA} + \frac{x}{R}} = \frac{1}{\frac{1}{1.1} + \frac{x}{0.34}}$$
$$x = 1.37 \text{ in.}$$

The author has pointed out in his paper that there are several factors which destroy the insulating property of foil. Dust and carbon deposits, condensation and tarnishing will raise the emissivity so that it approaches that of ordinary building materials. Hence, when the air space is divided into two parts by the foil the only insulating value obtaining, if the previously mentioned factors raise the emissivity of the foil, will be that of the additional air space formed by the sheet of foil.

It was observed by Dickinson and Van Dusen[8] that increasing the height of the air space up to about 24 in. increased the insulating value of the air space. Substantial evidence confirming this phenomenon was obtained and reported by the Engineering Experiment Station of The Pennsylvania State College.[7]

R. P. COOK: This paper is very timely in view of the increasing interest in the use of aluminum foil and other bright metallic surfaces as heat insulation.

I believe there is some hesitancy on the part of many engineers to accept this form of insulation because of the possibility of loss of effectiveness due to dulling of the bright surfaces. I wonder if Professor Rowley can give us any idea as to the decrease in insulation value which is likely to result from natural oxidation and tarnishing under ordinary conditions of exposure, such as within the walls of a house.

M. K. FAHNESTOCK: Mr. Close, in his discussion, states that one large manufacturer of domestic refrigerators has in the past year discontinued the use of aluminum-foil insulation in its boxes. From this discussion it is intimated that such action was probably based on the thermal insulating qualities of the material alone. I would like to suggest that some other factors, such as cost, might also have entered into the decision to discontinue the use of the foil insulation.

L. A. HARDING: The reduction in heat transmission, of the average insulated wall employed in residences, by the application of aluminum foil to one surface of the insulating board is approximately 13 per cent, as one may readily determine by comparing the calculated unit transmission u of the wall with and without the foil.

The heat transmission of the walls of the average residence (average of 400 residences) represents approximately 29.6 per cent of the total estimated heat loss of the building. (See page 548, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932.)

The estimated fuel saving by the use of aluminum foil, applied as indicated, is: 0.13×0.29 or approximately 4 per cent. Obviously, an increase in the thickness of insulation would accomplish the same result. If, however, the insulative board is installed to provide two air spaces instead of one, the heat transmission of the wall will be reduced to practically the same value as with the foil covered insulation.

In view of the fact that the original brightness of the aluminum foil surface must be maintained to effect the saving and that its reflecting power is readily destroyed by the accumulation of dust or the precipitation of moisture on the surface, there remains considerable doubt in the minds of some engineers as to its actual insulative value over a period of years in residence construction.

R. T. MILLER: In view of the publicity and advertising which bright metallic surfaces are receiving the writer believes that several points in Professor Rowley's paper are well taken, particularly in connection with the difficulties in making a theoretical

[7] Importance of Radiation in Heat Transfer Through Air Spaces, E. R. Queer, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 77.

[8] Testing Thermal Insulators, H. C. Dickinson and M. S. Van Dusen, A. S. R. E. Dec. 1915.

analysis, or in establishing absolute coefficients by test procedure, which can be indiscriminately applied to all cases. Very little has been said regarding the difficulties which a consumer might encounter, due to use of bright metallic surfaces. It is the writer's idea to bring a few of these difficulties to your attention.

All investigators and experimenters working on the tests of bright metallic surfaces will agree that the results obtained in their tests are only applicable under the same conditions as the tests, and that any change in brightness, any coating of film over the surface, will either greatly reduce, or entirely destroy their insulating value.

In Professor Rowley's paper, he points out the difficulty in preventing condensation from forming on the surface of the foil. This is substantiated by independent tests. A portion of this report is as follows: An inspection of this wall, No. 100, after test, indicated that there had been condensation on the foil, which left a slight fog or mist on the aluminum foil lining. Special tests with aluminum coated surfaces by the hot plate method showed that in several instances condensation occurred on the cold side, which materially reduced the effectiveness of the aluminum foil. In further tests, calcium chloride was placed in the air space, and practically corrected the difficulty. It is probable, therefore, that if the aluminum foil is placed on a surface, the temperature of which is below the dew point of the air within that surface, condensation will occur, and this condensation will materially effect the heat transmission of that surface.

It would seem to the writer that if it was necessary to place a dehydrating agent in the test wall, where it is possible to have almost ideal conditions, to obtain the insulating value of the bright metallic surface, it would be impossible to obtain duplicate results without the use of some dehydrating agent in the actual wall for ceiling construction.

In addition to the above, it is a well known fact that dust and dirt films on the face of bright metallic surfaces, either materially reduce the reflectivity or emissivity, as the case may be, if not entirely destroying both. The writer questions if it is possible to use bright metallic surfaces in building construction, and prevent the dust and dirt films from coating the surface of the metallic surface.

A very good example of what might be expected is found in any office or residence, by examining the glass in the windows, doors or partitions. It is necessary to wash and clean this glass periodically. It is impossible to clean metallic surfaces, or remove the dust and dirt film, when these surfaces are built into wall or ceiling construction.

Should anyone question the infiltration of dust and dirt into walls or roof, they need only examine any building that is being remodeled.

In Professor Rowley's paper, he states that he even finds a difference in the emissivity factors of material picked from different shipments, which might be due to oxidation or corrosion, or some effect on the brilliancy of the surface, other than dust and dirt, or condensation.

The writer also notices in going through the data on the tests of bright metallic surfaces, that most of the research work has been done with high mean temperatures. These temperatures hardly compare with the temperatures found in ordinary building construction.

The writer does not question but what there is a possible use for bright metallic surfaces in thermal insulation, but he does not believe that the field for this product is in house or building construction.

E. B. SVENSON [9]: It seems that discussion on Professor Rowley's paper has degenerated into a pogrom against aluminum foil insulation. Having been closely associated with the development of aluminum foil insulation in the United States, I feel that a few remarks on the other side would be in order.

There have been many successful applications of aluminum foil insulation, and we, therefore, should be careful that hasty conclusions do not prevent us from taking

[9] Aluminum Co. of America.

advantage of things that have real worth. It is unfortunate that there are no applications of aluminum foil insulation in the United States older than four years. In Europe there are applications several years older. It is, therefore, impossible to state, from a historical standpoint, what will happen to aluminum foil after 20 years of service.

Lloyds of London made certification of the condition of aluminum foil insulation after five years of service on the refrigerated spaces of a motor-ship operating in the tropics; that it was in as good condition as when installed. The engineer's log of operating time of the compressors indicated that the insulation had also maintained its original efficiency. There are few conditions more severe from the standpoint of condensation than those encountered in refrigerated equipment in the tropics.

The insulating value of reflective surfaces was discovered by students of heat transmission. The fact that aluminum is the most favorable metal for taking advantage of this phenomena is due solely to the properties of aluminum and not to any efforts of the manufacturers of aluminum.

All aluminum is covered with a thin layer of aluminum oxide which forms immediately on a fresh surface. If it were not for this fact, there would be no aluminum industry, since aluminum itself is one of the most active of the metals. The aluminum oxide is quite resistant chemically, in fact it is quite difficult to increase the natural thickness except by means of electric current and strong electrolytes. Aluminum foil is made from comparatively pure metal, and the oxide layer on the foil is, therefore, continuous and affords excellent protection to the metal underneath.

Aluminum foil does not tarnish and is quite resistant to the compounds of sulphur which seriously attack common structural metals.

There is a further fact which must not be lost sight of with regard to the use of aluminum foil as insulation. Aluminum foil is impermeable to air, vapors, water, etc., and its use in buildings serves to wind proof and moisture proof those buildings and thereby keep the infiltration of dust or other dirt to a very small amount.

Professor Harding has remarked that in one instance where they computed the heat saving by the use of one layer of foil that it amounted to only 4 per cent of the total. Such cases are possible with other insulations as well as aluminum foil; however, the effect of a single layer of foil on one side of the air spaces in buildings is equivalent to the addition of ½ in. of the common forms of insulating material in the same position. This is a common method of using the board forms of insulating material, and I doubt if the builders and manufacturers interested in such applications would agree that they were not worthwhile.

Aluminum foil insulation was not discovered by the manufacturers of aluminum; however, before they could go on record as to its efficiency and serviceability, it was necessary to check the available information. This was done at considerable expense, but as a result of tests conducted by the manufacturers of aluminum foil, its value for insulating purposes has been proved, and today, aluminum foil is giving extremely satisfactory service under many different types of insulation conditions.

F. B. ROWLEY: In general this discussion seems to have centered around the permanency which may be expected for the insulating value of the foil. Since the insulating value is due to the low emissivity coefficient of the material there is no question but that it must be maintained clean and bright if it is to be efficient. The question as to what will happen to the surface over a period of years when installed in the air space of the wall is one which must be determined by a further investigation and experience. In order to get the insulating value of the foil it must be used with discretion, and it cannot be considered as a substitute for all other forms of insulation. There are certain places where bright metallic surfaces may be used to an advantage and others where it would be useless to apply them and expect any results. In applying any insulation an engineer must recognize its good qualities and limitations and not expect one type to fulfill all requirements. There is always the chance that, with a new product, enthusiasm may outweigh good judgment in its application.

No. 997

HEAT TRANSFER FROM DIRECT AND EXTENDED SURFACES WITH FORCED AIR CIRCULATION

By G. L. Tuve * *(MEMBER)* and C. A. McKeeman † *(NON-MEMBER)*
Cleveland, Ohio

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers and conducted at Case School of Applied Science

I. Introduction and Summary

HOW does the heat transfer from a forced-convection air heater vary with the air velocity? Is the heat transfer coefficient (per degree of temperature difference) the same for either air heating or air cooling? Are the commercial fin-tube units as effective per square foot of air-side surface as are plain pipe or cast iron sections? What is the best arrangement for securing high heat transfer with a minimum of pressure-drop?

These and many other questions occur to the engineer and should be answered by experimental and research work. While much data are already available, many questions remain to be answered before engineers can predict with confidence the performance of their designs. This paper is a preliminary report on a project in this field, which has been undertaken by the Mechanical Engineering Laboratories of Case School of Applied Science, with the cooperation of the Committee on Research, of the A. S. H. V. E. A study and correlation of available data from the literature are here presented, and experimental results on certain phases of the problem are given.

1. The *major* factors affecting the heat transfer coefficient of finned tubing (or other air heating or cooling units) are three: (a) The design and surface-arrangement of the unit. (b) The mean velocity of the air stream. (c) The character of the air stream, with particular reference to the presence of local eddies or local turbulence. (Surface condensation and evaporation are another major factor in the case of air coolers, but these merit separate study and are not being considered in this paper.)

2. Of *lesser* importance are the diameter of the tubes, the temperature, the kind and the velocity of heating or cooling fluid used in the tubes, and the direction of heat flow, *i.e.*, air-heating or cooling.

* Associate Professor of Heat-Power Engineering, Case School of Applied Science.
† Assistant Professor of Mechanical Engineering, Case School of Applied Science.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buck Hill Falls, Pa., June 1934, by C. A. McKeeman.

3. It is possible to calculate the heat transfer coefficient by a simple equation, but a few tests must be made on each new design-arrangement in order to establish the constants in the equation.

4. The use of flow-disturbers in the air stream immediately preceding a fin-tube section, may increase the heat transfer by 50 per cent or more in certain cases.

5. High velocities are somewhat less effective with extended surface units than with plain tubing, and it is *not* safe to assume that the heat transfer varies as the 0.8 power of the velocity, as it may not vary at a rate greater than the square root of the velocity.

6. There is a definite need for further research in this field (See Sec. IV, *Conclusion*).

II. HEAT TRANSFER EQUATIONS

In order to arrive at a method for calculating the heat transfer of a forced-convection air-heating (or cooling) unit,[1] the following variables must be considered:

1. Mean difference in temperature between the air stream and the hot surfaces.
2. The area, shape, size and surface arrangement of the hot surfaces.
3. The mean velocity of the air stream.
4. Character of the air stream as regards turbulence and local eddies.
5. The absolute temperature of the air, with particular reference to the air film next to the heating surfaces.
6. The character of the heating fluid used in the tubes, its velocity, etc.
7. The effectiveness of the extended surface (if any), compared with the prime or direct heat-transfer surface.

Since the surface resistance to heat flow on the outer or air-side is much greater than the similar resistance on the inside of the pipe, it is desirable to compensate for this by increasing the area of the air-side. The simplest extended surface arrangement consists of circular fins of uniform thickness attached to standard round tubing. The effectiveness-factor for the fin surface as compared with the tube surface then depends upon the thickness of fins, depth of fins, distance between fins, and the method of attaching the fins to the prime surface.

It is most convenient to calculate the total heat transfer by the usual simple equation:

$$q = A\,U\,(T_1 - T_2) \tag{1}$$

where

q = heat transfer in Btu per hour
A = area of air-side surface, square feet
U = overall coefficient of heat transfer per square foot of air-side or external surface, per degree Fahrenheit temperature difference.
$T_1 - T_2$ = mean difference in temperature between the heating fluid and the air. Usually the logarithmic mean temperature difference should be used.

---

[1] In most of the discussion which follows, an air-heating unit is assumed for convenience. The data and the methods outlined apply as well to air coolers, however, as long as no surface condensation is involved.

The overall coefficient U may be calculated by combining the inside and outside film conductances and the conductance of the wall, but since the conductivity of the metal wall is high and its thickness is small, it may be left out of consideration with but little effect on the degree of accuracy. Then:

$$U = \frac{1}{\frac{R}{h_1} + \frac{1}{h_2}} \tag{2}$$

where h_1 is the hot-side and h_2 the air-side film or surface coefficient, and R is the ratio of air-side surface to hot-side surface. This equation is sufficiently accurate for commercial pipe or tubing, but does *not* account for temperature gradients in extended surfaces.

The general equation for the surface coefficient in forced convection, which may be derived by dimensional analysis, is as follows:

$$\frac{hD}{k} = b\left(\frac{DG}{\mu}\right)^n \left(\frac{c\mu}{k}\right)^m = b\, R^n S^m \tag{3}$$

where all quantities are expressed in dimensionally consistent units (usually foot—pound—hour), and the symbols are defined as follows:

$\frac{hD}{k}$ = Nusselt number (dimensionless).

$R = \frac{DG}{\mu}$ = Reynolds number (dimensionless).

$S = \frac{c\mu}{k}$ = Stanton number or Prandtl number (dimensionless).

h = film coefficient or surface conductance.
D = diameter (or other linear dimension), feet.
G = mass velocity = linear velocity × density.
c, μ and k are respectively the specific heat at constant pressure, the absolute viscosity and the thermal conductivity of the fluid.
b is the proportionality constant.

This equation, usually ascribed to Professor Nusselt, has been accepted by the leading authorities on heat transfer, and will be found in the standard books in various languages (such as those of McAdams, Schack, Fishenden and Saunders, Grober, etc.). This form of equation should be used when a high degree of accuracy is essential.

For the forced convection of air, the Nusselt equation may usually be simplified, since for moderate ranges of temperature the Stanton number is constant, and the individual values of air conductivity, viscosity and specific heat may even be taken as constant. If this is done, the simplified Nusselt equation becomes:

$$h_2 = \frac{BG^n}{D^{1-n}} \tag{4}$$

where G is the mass velocity, in pounds per hour per square foot and D the diameter in feet. The values of B and n in this equation must be determined by experiment.

For extended-surface units the actual use of Equation 4 is attended with great difficulty because of the temperature gradient of the air-side surface

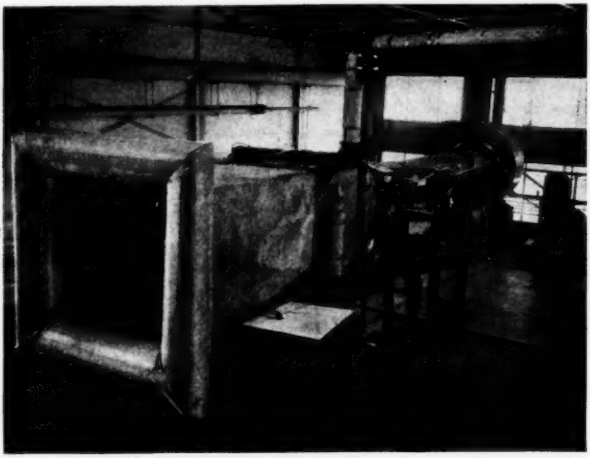


FIG. 1. GENERAL VIEW OF TEST DUCT FROM INLET END

from the base to the periphery of the fin, and because both fin temperatures and adjacent air temperatures are very difficult to measure.

For practical purposes, therefore, these temperature gradients and the rela-

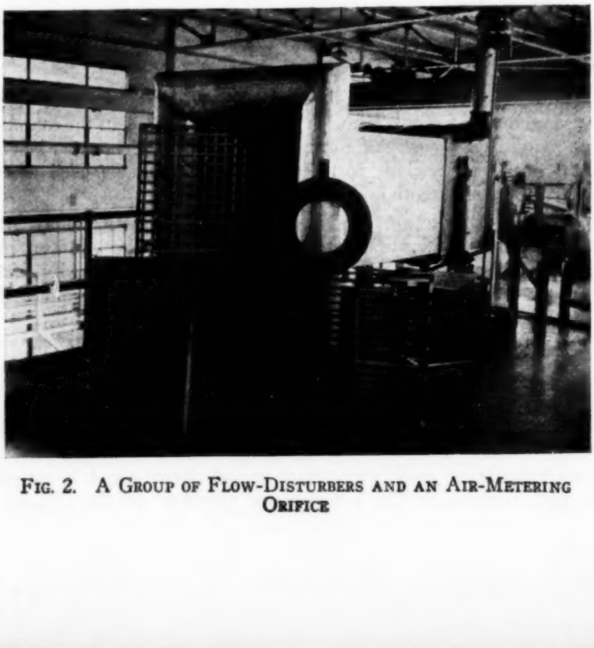


FIG. 2. A GROUP OF FLOW-DISTURBERS AND AN AIR-METERING ORIFICE

tions defined by Equation 2 as well are taken into account by writing a new equation for the overall coefficient, in the form:

$$U = \frac{BG^n}{D^{1-n}} \tag{5}$$

The justification of this form of equation rests entirely on its convenience and on the fact that experimental results are consistent with it.

The values of exponent n and constant B in Equation 5 will *not* be the same as the corresponding values in Equation 4, even if no appreciable temperature gradient exists along the fins. The deviation between the two values of exponent n will not be great, but the exponent of Equation 5 will be smaller.

Recommended Calculation Procedure

In view of the fact that comprehensive experimental data justifying calculation refinements are not available, the authors recommend that until such data are obtained all forced-convection air heater and air cooler calculations be based on the simple equations (1) and (5), or a combination of the two in the form:

$$q = \frac{ABG^n}{D^{1-n}} \quad (T_1 - T_2) \tag{6}$$

III. EXPERIMENTAL WORK AT CASE SCHOOL OF APPLIED SCIENCE

Evolution of Test Equipment

Work has been done at Case School of Applied Science on five different projects, involving as many test arrangements. Both air heating and cooling have been studied, using hot and cold water and steam as heat carriers, and both plain and finned units have been tested. Blow-through and draw-through arrangements have both been used. While several of the variables already mentioned have been studied at some length, the most significant result has been the demonstration of the importance of air-stream conditions, and the results presented herewith are confined to this phase of the subject.

When this work was started, there was no intention of making any special study of air-flow conditions, but it was soon found that these conditions were so important that their effects masked completely the effects produced by certain other variables which the study was to cover.

The first test unit was attached to an air conditioning system, and was tested both as an air heater and an air cooler, using hot and cold water respectively as a circulating medium. With this set-up the variations in heat transfer coefficient with mean air velocity were not as consistent as was desired, and this was found to be due to an unequal distribution of velocities across the face of the test section, accompanied also by changes in the amount of turbulence or local eddying in the air stream as the air velocity was varied. An unequal velocity distribution results in heat transfer coefficients which are low compared with those for uniform velocity across the face of the unit, while local eddies or turbulence result in high coefficients.

To obtain a uniform velocity across the entire test section and an undisturbed or smooth flow, a draw-through unit with a large inlet-bell was decided upon, and four different arrangements were tried, the most satisfactory being that

shown in Fig. 1. In these units the air flow was regulated entirely by a variable-speed fan with infinite step rheostats, and to determine the velocity- and temperature-distribution on the upstream side of the test section, traverses were made with an impact tube and micro-manometer and with a thermocouple, respectively. With this arrangement, almost no variation of velocity or air-inlet temperature could be detected.

Tests were next confined to air heating by steam, using first a series-flow arrangement in horizontal tubes, and later a parallel-flow unit with vertical tubes. In all cases, only the tubes projected into the air stream, the return

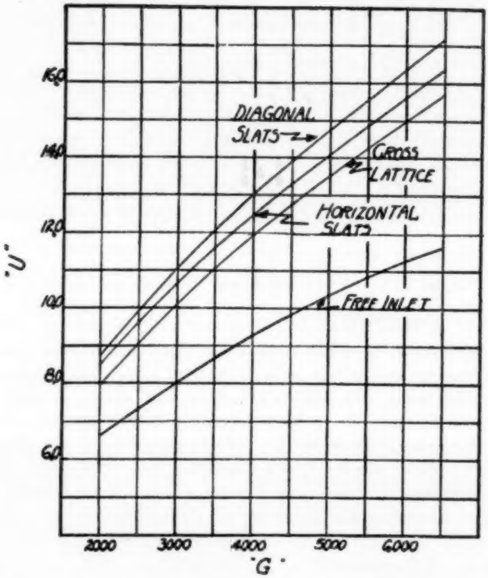


FIG. 3. EFFECT OF FLOW-DISTURBERS OF VARIATION OF OVERALL COEFFICIENT U WITH MASS-VELOCITY, G

bends or headers being mounted in insulated boxes outside the duct. Low-pressure steam was supplied from either a reducing-valve or a gas-fired boiler with accurate pressure control, and was passed through an electric superheater before it entered the test section. The condensate drain was automatically controlled by using a U-tube waterleg or gravity seal. Continuous vents of ample size were attached at both ends of the condensate header.

In the test runs, after allowing ample time for conditions to become steady, readings were taken every three minutes, during one-half to one hour. Condensate weights were read to 0.01 lb and temperatures to 0.1 F. Air quantities by heat balance were checked by measuring the air flow with a calibrated orifice, and the agreement was within 5 per cent for all runs. Both thermocouples and thermometers were tried for air temperature measurement, but the

thermometers, if shielded from radiation, were found to be just as accurate and more convenient. Four readings each of inlet- and outlet-air temperatures were used, but the variation was very small because special mixing devices were used on the outlet side to prevent stratification. The entire test duct and all pipe connections were well insulated, but heat losses were approximated and corrections made therefor. Corrections for radiation from the test section were calculated, but were found to be negligible.

One test unit was constructed in which an elaborate series of fin-temperatures

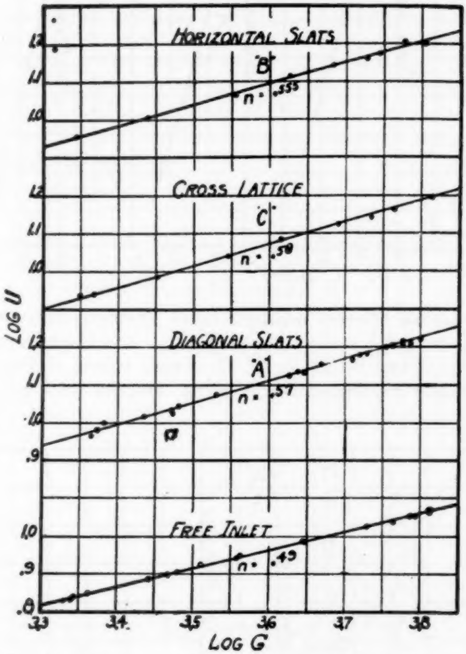


FIG. 4. CURVES OF FIG. 3 PLOTTED ON LOGARITHMIC SCALES, WITH TEST POINTS AND SLOPES INDICATED

were measured, but more data on fin-temperature gradients is to be obtained, and this phase of the subject will be reserved for a later paper.

Nature of Tests

The tests here reported (Figs. 3 to 6 inclusive), were all made for the purpose of determining the effect of two variables: (1) Mean air velocity, and (2) Air disturbances or local eddy-velocities. Linear velocities from 300 to 1,500 fpm through the free area were covered. Fig. 2 shows some of the grids or flow disturbers which were used in the inlet duct in front of the test section.

These tests were all made on a 35 in. square test section in a single row

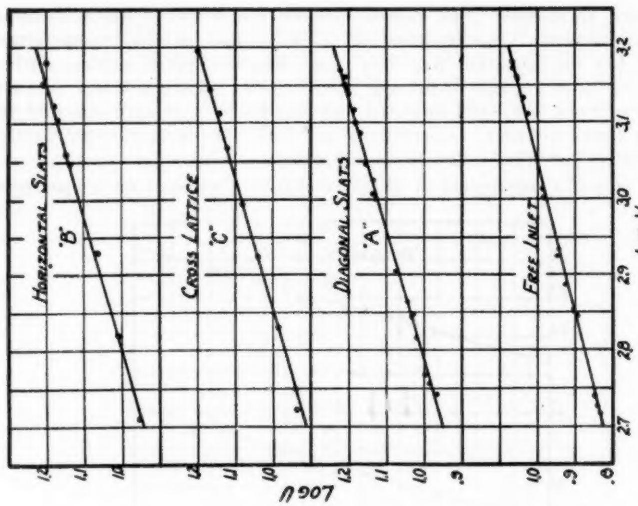


FIG. 6. CURVES OF FIG. 5 PLOTTED ON LOGARITHMIC SCALES, WITH TEST POINTS INDICATED

with ¼ in. clearance between adjacent tubes. Overall diameter of the finned tubing was 1⅜ in. and it was constructed of ⅜ in. smooth copper ribbon 0.012 in. thick spirally wound on ⅝ in. O. D. copper tubing, 7 fins per inch. The assembled unit was tin-dipped for bonding, and its total weight was 0.7 lb per foot.

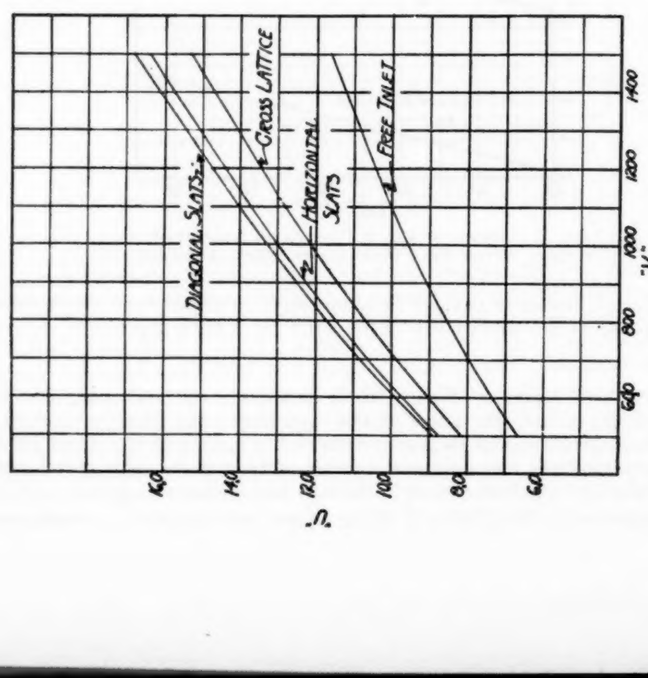


FIG. 5. EFFECT OF FLOW DISTURBERS ON VARIATION OF OVERALL COEFFICIENT U WITH MEAN LINEAR AIR VELOCITY V

Tests were made with several types of disturbers or grids having free areas from 25 per cent to 90 per cent of the duct area and spaced at various distances from ½ in. to 8 ft from the upstream face of the test section.

Test Results and Conclusions

In Fig. 3 are shown the values of the overall coefficient U, (Equation 1), plotted against the mass-velocity of the air G, for three types of flow disturbers or grids, and for free-open inlet or undisturbed flow. The same curves with the test points indicated, are shown separately in Fig. 4. Fig. 5 gives the same data plotted against linear air velocity V, and these curves are shown separately in Fig. 6. All velocities represent the mean air speed through the free or unobstructed area between fins and tubes, *i.e.*, the area considered is the duct area less the projected area of the finned tubing. In these tests the disturbers were mounted one inch in front of the inlet face of the test section.

Fig. 7 shows the pressure-drop losses through the test section including the loss through the disturber-grid.

From these curves and from the supplementary tests, the following conclusions are indicated:

1. For this particular finned tubing the heat transfer coefficient for undisturbed air flow at uniform velocity varies approximately as the square root of the air velocity, as represented by the equation:

$$U = 0.052\frac{G^{0.43}}{D^{0.51}}$$

2. By producing local eddy-velocities near the test section this heat transfer may be increased, and the increase is greater at high velocities. This is indicated by the equation applying to disturber-grid A:

$$U = 0.045\frac{G^{0.57}}{D^{0.43}}$$

3. For producing a moderate increase in the capacity of an extended-surface unit, the disturber method may involve a lower first cost than either increasing the size of the unit or increasing the air velocity. The method should be useful where air velocities are limited by other conditions, and where a compact unit is desired. The pressure-drop produced by the grid need not be excessive, as shown in Fig. 7.

4. A disturber-grid of about 50 per cent free area is recommended. If the area is further reduced, the pressure-drop increases much more rapidly than the heat transfer. The shape of the elements composing the grids is of less importance. Narrow flat strips set at a 45-deg angle were found to be the most effective. Nozzle-shaped air passages showed little if any reduction in the friction-drop. For best results the air disturber should be placed from ½ in. to 1½ in. from the face of the heat-transfer unit.

Precautions to Be Observed in Using Fin-Tube Units

The following practical rules are suggested to cover the operating conditions of extended-surface heaters and coolers:

1. Proper distribution of the heating or cooling fluid is very important on

TABLE 1. SUMMARY OF EQUATIONS FOR HEATING AND COOLING OF AIR BY FORCED CONVECTION. (SHOWING EFFECTS OF AIR VELOCITY AND PIPE DIAMETER)

Item No.	Equations, Experimenters and Applications (G = pounds/square foot/hour; D = feet)	Heat Transfer Coefficients for Heating Air at 100 F with Pipe Diameters and Velocities of:			
		Diam 1 in. V=500 fpm	Diam 1 in. V=1500 fpm	Diam 2 in. V=500 fpm	Diam 2 in. V=1500 fpm
1	$h = 0.0035 \dfrac{G^{0.8}}{D^{0.2}}$ Nusselt, Josse, Royds & Campbell. For flow inside ¾ in. to 1 in. smooth pipes.	2.6	6.4	2.3	5.6
2	$h = 0.0085\ G^{0.78}$ Jurges, Taylor and Rehbock. For parallel flow over 6 in. to 18 in. square, smooth, flat plates.	(No effect of diameter)		1.8	4.1
3	$h = 0.0183 \dfrac{G^{0.69}}{D^{0.31}}$ Reiher, Carrier & Busey, Rietschel. For flow over several rows ⅝ in. to 1¼ in. smooth tubes, staggered.	7.8	16.8	6.4	13.7
4	$U = 0.049 \dfrac{G^{0.585}}{D^{0.415}}$ Commercial bulletin on fin-tubes. Flow over 1 row 1⅜ in. O. D. fin-tubes, ⅜ in. fins, 7 per inch.	12.2	23.1	8.5	17.7
5	$U = 0.045 \dfrac{G^{0.57}}{D^{0.43}}$ Authors, Case School Research. Flow over 1 row 1⅜ in. O. D. fin-tubes, ⅜ in. fins, 7 per inch, with flow disturber of diagonal slats.	10.4	19.4	7.7	14.3
6	$h = 0.033 \dfrac{G^{0.56}}{D^{0.44}}$ Hughes, Reiher, Vornehm, Gibson. Flow over single tubes or pipes. ¼ in. to 2 in. O. D.	6.7	13.3	5.1	10.0
7	$U = 0.052 \dfrac{G^{0.49}}{D^{0.51}}$ Authors, Case School Research. Flow over 1 row 1⅜ in. O. D. fin-tubes, ⅜ in. fins, 7 per inch, with smooth flow-stream.	7.9	13.4	5.5	9.4
8	$h = 0.061 \dfrac{G^{0.51}}{D^{0.49}}$ T. E. Schmidt[a]. Flow over 6 rows 7 in. O. D. fin-tubes, 2 in. fins, 15 per foot.	9.0	15.7	7.3	12.7
9	$h = 0.044 \dfrac{G^{0.4}}{D^{0.6}}$ L. V. King. Flow over very small wires	100 for diameter 0.005 in. and 500 fpm velocity			

[a] For curves showing Schmidt's results, see article by King and Knaus, *Mechanical Engineering*, May, 1934. This article also gives other curves showing similar data to above table.

account of the high capacity of the fin tubes per foot of tube length. This calls for high velocities, if a liquid is used as the heat carrier, and adequate venting if steam is used. A combination of series and parallel flow is a good arrangement in either case, but if the tubes are operated in parallel with steam, distribution-orifices at the tube inlets may be necessary. In any case the distribution should be checked by feeling the temperature of the fins, if no more accurate method is available.

2. The discharge air from such a unit is highly stratified, and attempts should not be made to measure its temperature without thorough mixing, and even then four or more shielded thermometers should be used.

3. Duct and heat-transfer surface arrangements should be such as to increase

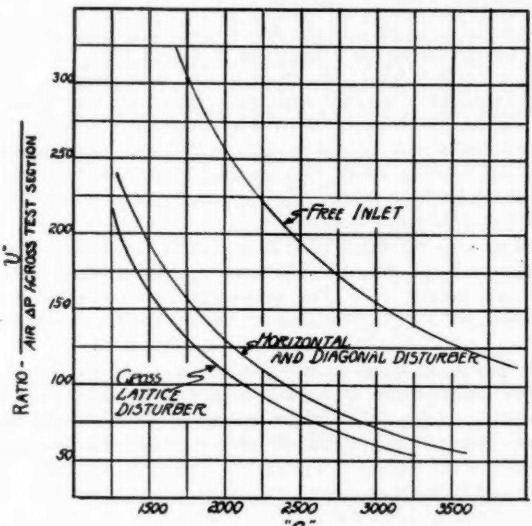


FIG. 7. RELATION BETWEEN HEAT TRANSFER AND PRESSURE-DROP. ORDINATES ARE BTU PER HOUR PER SQUARE FOOT PER DEGREE DIFFERENCE BETWEEN STEAM AND AIR, PER INCH OF PRESSURE-DROP THROUGH TEST SECTION AND DISTURBER GRID

the local turbulence as much as possible, though great inequalities in the velocity distribution across the face of the test section are undesirable.

IV. CORRELATION WITH PREVIOUS INVESTIGATIONS

The main value in experimental results such as those reported above, lies in the extent to which they may supplement or extend the data already available on the subject. A variety of data have been published on the heating and cooling of air by both direct and extended surfaces, but the number and character of the variables involved is such that the effect of each cannot easily be evalu-

ated. As a summary of the available data, Table 1 has been prepared, using Equations 4 and 5 as a basis of analysis.[2] This table gives values for the constants B and n for the calculation of surface coefficients by Equation 4, and overall coefficients by Equation 5. It also gives examples of heat transfer coefficients calculated by each equation, for two air velocities and two pipe diameters. These examples are for heating air by low-pressure steam (the mean air temperature being assumed at 100 F for density calculations).

From Table 1 it is again apparent that the major variables are: (1) Mean air velocity, (2) Air-stream disturbances or local turbulence, and (3) Nature and arrangement of surfaces. These factors, and a fourth factor of temperature level and temperature gradients, will now be discussed in turn.

Air Velocity and the Velocity Exponent n

The value of the exponent n in the heat transfer equations for either surface coefficient or overall coefficient (Equations 3, 4, and 5,) is in the case of any given section a direct index of the effect of air velocity on the heat transfer. Table 1 shows that this exponent varies from 0.4 to 0.8, and Table 2 has been prepared to indicate the importance of this variation in practical calculations. Thus if the heat transfer is 10 at a velocity of 200 fpm, and it varies as the 0.4 power of the air velocity, it will have reached 25 at 2000 fpm, but if the heat transfer varies as the 0.8 power of the air velocity, it will increase from 10 at 200 fpm to 63 at 2000 fpm.

In any specific case the value of n is dependent on three things: (1) The diameter, shape and smoothness of the heat-transfer surface. (2) The character of the air stream. (3) The temperature-gradients along the fins or extended surface, if any. (This last statement applies to Equations 5 and 6 only.)

As regards the actual value of the velocity exponent for finned tubing and other extended surface units, McAdams states that it varies from 0.58 to 0.74 (*Heat Transmission*, page 233) and an analysis of the performance tables published by five American manufacturers shows values covering approximately this range.

Effect of Air-Stream Disturbance

Unfortunately we have no satisfactory means for observing or measuring the smoothness of an air stream. We may state that Reynolds number is a criterion of turbulence, but this tells only a small part of the story. Noise furnishes some index of stream conditions, but there are many other sources of noise in a system. Photographs of smoke flowing in the air stream are illuminating but they are difficult to get and are successful only over a small range of velocities. Friction or pressure-drop furnishes a good index, but only in case the system remains geometrically similar. The hot-wire anemometer has possibilities for turbulence measurement, but this application is only in an experimental stage.

It thus becomes necessary to make a mere estimate of the degree of flow disturbance from a knowledge of the operating conditions. Departures from smooth-stream conditions are a function of the geometrical shape and surface smoothness of the objects or walls over which a fluid is flowing. Local eddies

[2] In view of the excellent bibliography arranged alphabetically in McAdams, *Heat Transmission*, authors' names only are given in this paper and the reader is referred to the above-mentioned bibliography for sources.

or vortices may be created, and persist in the stream for some distance past the point of their formation. Apparently anything which will assist the moving air in scrubbing off the viscous or stagnant air film adhering to the heating surface, increases the heat transfer by bringing a larger percentage of the total air particles into close proximity with the hot surfaces. In many cases of stream disturbance, high local velocities exist, and these may play a major part in increasing surface conductance.

Most efforts to produce disturbed air conditions have accomplished this by the shape and arrangement of the heat transfer surfaces with respect to the air stream. Reiher's data on 5 rows of plain 1-in. pipes showed about 28 per cent greater heat transfer for the staggered arrangement than for the in-line arrangement. Similar results were reported earlier by Carrier and Busey. Rowley, Jurges, Ott and others have reported increases in surface coefficients of 5 per cent to 30 per cent with rough plane surfaces as compared to smooth ones. Rowley found that for a flat plate mounted in a wind of 15 to 25 mph velocity, the surface coefficient was increased in varying amounts, up to about 25 per cent as the plate was placed at various angles with the stream from 0 deg to 90

TABLE 2. EFFECT OF THE VELOCITY EXPONENT n

(See Equations 4, 5 and 6)

Values of Exponent n	Relative Heat Transfer (Surface Conductance) for Velocities of 200 fpm	500 fpm	2000 fpm
n = 0.4	10	14.4	25.1
n = 0.5	10	15.8	31.6
n = 0.6	10	17.3	39.8
n = 0.7	10	19.0	50.0
n = 0.8	10	20.8	63.0

deg. Even larger increases were reported by Reiher. Schey and Bierman obtained an increase of about 50 per cent in the heat transfer of a finned airplane engine cylinder at flying speeds, when it was placed at 45 deg with the air stream as compared with either the parallel or the 90 deg positions. Reiher and Hughes both demonstrated that the heat transfer from a cylinder in an air stream is increased by placing strips or tubes in front of it.

Corrugated-plate heat exchangers have shown a larger coefficient than flat-plate exchangers. Baffles and helical guide-shapes have been used inside pipes to increase heat transfer. Many types of baffles are used in shell-and-tube exchangers for liquids. Screens and decorative grilles are used in front of automotive radiators.

But it is difficult to classify these flow disturbers and little has been published regarding the quantitative evaluation of their effects at various air velocities. There is no doubt but that such disturbers are more effective at high fluid velocities than at low ones. The results shown in Fig. 5 are conclusive on this point.

Friction loss or pressure drop is important in determining the power required to move the air stream and hence the relation between increase of heat transfer and increase in friction loss is an important one. In the case of two disturbers which have the same effect on the heat transfer, the one may produce much less friction loss than the other. Colburn and King demonstrated that by breaking up the flow of air through a tube by packing it with granular material, the

heat transfer could be increased as much as six times, but the friction loss was thereby increased 200 times. They concluded that the coefficient might to better advantage be increased by using a higher velocity in a plain tube.

Friction losses are of two kinds, skin friction and eddies or turbulent friction. While much of the data given in this paper indicate a close relation between

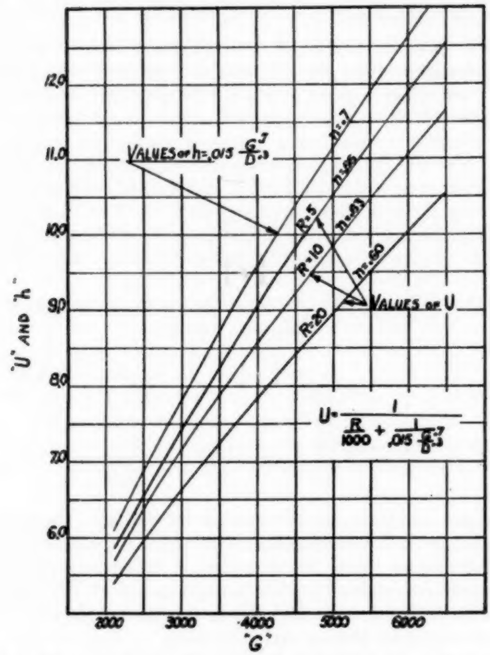


FIG. 8. VARIATION OF OVERALL COEFFICIENT OF A FIN-TUBE UNIT WITH RATIO OF AIR-SIDE SURFACE TO STEAM-SIDE SURFACE; NO TEMPERATURE GRADIENT IN FINS

local turbulence and heat transfer, there is also evidence that the latter is related to skin friction. Hughes found greater heat transfer per unit area for a streamline section than for a cylinder, and from this Fishenden and Saunders conclude, "it appears that heat transfer is directly related only to skin friction."

Effect of Temperature

Temperature effects are of two kinds, those due to the absolute temperature level and those due to temperature differences or temperature gradients.

The surface coefficients are affected somewhat by the absolute temperature since the viscosity (μ), the conductivity (k) and the specific heat (c) of the fluid are affected (see Equation 3). To illustrate the magnitude of these effects, examples of typical cases of air heating and air cooling are included in Table 3 (see *Conclusion*).

TABLE 3. EXAMPLE OF DIFFERENCES DUE TO TEMPERATURE LEVEL

(Calculated for 1½-in. O.D. fin-tube unit, same unit in all cases.)

Temperature Conditions	Air-Surface Heat Transfer Coefficients at Velocities of— 500 fpm	1500 fpm
HEATING: Typical air-heater using steam at 5 lb gage....	7.0	13.5
COOLING: Typical air-cooler using high-velocity water circulation at 40 F..................................	6.5	12.5

While the difference due to temperature level is large enough to be worth taking into account in accurate heat transfer calculations, it is smaller than the uncertainty which usually exists regarding the air-stream conditions.

The effect of temperature differences and temperature gradients is discussed in the next section.

Effect of Nature and Arrangement of Surfaces

From Equation 3 the effect of tube diameter is shown to be proportional to D^{1-n}, and this relation has been confirmed by experiment for plain pipes and tubes. There are little or no data on this point with respect to finned tubing. The constants given in items 5, 7 and 8 of Table 1 have been derived on the assumption that the diameter should enter in the above manner for finned tubing also, and that the effective diameter is the overall diameter.

The effect of surface arrangement on the value of the velocity exponent n (because such arrangement affects the nature of the air flow), has been discussed, but the presence of extended surfaces affects the value of n in two other important ways. With thin fins, an increase of air velocity tends to bring the temperature of the fin periphery closer to the air temperature, and to set up a temperature gradient or difference between the prime surface and the extended surface. This makes the extended surface less effective and reduces the value of n in the equation for overall coefficient (Equation 5 or Equation 6).

But even if there is no temperature gradient in the fins, the mere existence of extended surfaces changes the value of n, as may be proven by substituting values in Equation 2. Fig. 8 shows this effect. Taking as an example a steam-heated fin tube with a 10 to 1 ratio of air-side to steam-side surface, (*i.e.*, $R = 10$), assume that the steam-side coefficient $h_1 = 1000$, and that the air-side surface coefficient is represented by the equation: $h_2 = 0.015 \frac{G^{0.7}}{D^{0.3}}$. Calculating values of U by Equation 2 and plotting them in turn against G, we have a new curve (Fig. 8) which indicates that U varies approximately as $G^{0.63}$. Other area ratios (values of R) give the other curves of Fig. 8, all other assumptions remaining the same.

Conclusion

Thus it is evident that while convenient methods of calculation are available after the constants for a particular design have been established by experiment, yet much remains to be learned about the effect of such primary variables as tube and fin diameters, fin thickness and fin spacing. We know a little about the practical effect of air-flow conditions, but we lack any method of ade-

quately measuring these conditions in a given installation and the theory or principles underlying these effects have not been clarified. We may state that the heat transfer coefficient varies directly with some power of the velocity, but whether this variation is as the square root or as the 0.8 power, or some value between, we can only predict after further work has been done on the effect of the primary design proportions. A rough classification of air heaters

TABLE 4. RECOMMENDED HEAT TRANSFER EQUATIONS FOR APPROXIMATE CALCULATIONS

Applications	Equation
Class I.—Large plain tubes, staggered arrangement or with flow disturbers.	$h = 0.022 \frac{G^{0.7}}{D^{0.3}}$
Class II.—Common large fin-tubes, staggered arrangement or with flow disturbers.	$h = 0.04 \frac{G^{0.6}}{D^{0.4}}$
Class III.—Small tubes with large fins, or small plain tubes widely spaced; smooth air flow.	$h = 0.05 \sqrt{\frac{G}{D}}$

Equations apply to both air heaters and coolers. h = Btu per hour per square foot of total air-side surface per degree difference between mean air temperature and primary tube-surface temperature. D is in feet. G is mass velocity in pounds per square foot per hour.

and coolers, with suggested heat-transfer equations and examples of the results obtained by their use are given in Table 4, but these values should be used only for approximations. For accurate results, a few test-runs on each design of unit are necessary.

DISCUSSION

M. K. FAHNESTOCK: You have stated that the relative humidity of the air used during your investigation was maintained approximately the same during each test. I would like to ask if any preliminary tests were run with air of unusually low or high relative humidities to determine if relative humidity had any effect upon the heat transmission.

PROF. MCKEEMAN: There were no tests run with air having extremely high or low humidities.

PROFESSOR FAHNESTOCK: We have just finished some tests on a gravity convector where the relative humidity of the air was varied for different tests between 5 and 68 per cent. There seemed to be a slight indication that in the case of a gravity convector the heat output decreased a very small amount with the higher relative humidities. I would like to suggest that several tests be made with unusually high relative humidities to see if this trend is more pronounced when using forced air.

F. B. ROWLEY: This paper is the result of well planned research work and is worthy of very careful study. It deals with the fundamentals in the problem of heat transfer from finned tubes. The authors have made an excellent analysis of the variables involved and show the difficulties in arriving at a simple solution of the problem. They have shown very clearly the effect of turbulent air flow on the surface coefficients and, while they indicate that more experimental work is necessary, they have brought their present results to a point of practical application.

Professor Tuve is a member of the Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation and has been largely responsible for the program of the Committee's work. It is to be hoped that the research work may be continued under Professor Tuve's direction at Case School of Applied Science.

No. 998

FACTORS AFFECTING THE HEAT OUTPUT OF CONVECTORS

By A. P. Kratz,* M. K. Fahnestock ** and E. L. Broderick,† (*MEMBERS*)
Urbana, Ill.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers and conducted at the University of Illinois

The data presented in this paper were obtained in connection with an investigation carried on in the Department of Mechanical Engineering as part of the work conducted by the Engineering Experiment Station of the University of Illinois. The investigation was conducted under the direction of A. C. Willard, Acting Director of the Engineering Experiment Station and Head of the Department of Mechanical Engineering. This paper includes results from the work constituting a continuation of the study of the performance of convectors and the material will ultimately comprise part of a bulletin of the Engineering Experiment Station.

TWO previous papers [1, 2] on this subject were devoted to discussions of tests run in a warm wall testing booth for the purpose of deciding the validity of the correction factor, applied in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) [3] for reducing the heat output obtained under test conditions to the equivalent heat output under standard conditions with steam in the heating unit at 215 F and a temperature of 65 F for the air at inlet. This correction factor is:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.3} \qquad (1)$$

where 150 = the temperature difference between steam at 215 F and inlet air at 65 F
t_s = the temperature of steam during the test, degrees Fahrenheit
t_1 = the average inlet air temperature during the test, degrees Fahrenheit

* Research Professor in Mechanical Engineering, University of Illinois.
** Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
† Research Assistant in Mechanical Engineering, University of Illinois.
[1] Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, A. S. H. V. E. Transactions, Vol. 38, 1932, p. 511.
[2] Tests of Convectors in a Warm Wall Testing Booth, Part II, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick, A. S. H. V. E. Transactions, Vol. 39, 1933, p. 319.
[3] A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), A. S. H. V. E. Transactions, Vol. 37, 1931, p. 367.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buck Hill Falls, Pa., June 1934, by M. K. Fahnestock.

The results [1, 2] in general indicated a reasonably close agreement between the heat output actually obtained with steam at 215 F and inlet air at 65 F and that calculated from the heat output at some other inlet air temperature through the medium of the correction factor, provided that the inlet air temperature did not exceed 80 F or fall below 60 F. Deviations as great as 4.5 per cent were observed, however, and it was recommended that the testing range be confined to inlet air temperatures between the limits of 60 F and 75 F. There seemed to be some indication that the percentage deviation between the actual and calculated heat outputs might be affected by the physical dimensions of the convectors, particularly by the height of the cabinet.

No analysis was made to determine whether the variation in heat output with the difference in temperature between the steam and inlet air for all of the types of convectors tested could be represented by an equation of the form:

$$H = K\ (t_s - t_1)^n \tag{2}$$

where H = heat output, Btu per hour
K = a constant
t_s = the temperature of the steam, degrees Fahrenheit
t_1 = the inlet air temperature, degrees Fahrenheit
n = a numerical exponent

and whether the correction formula should assume the form:

$$_cH_{150} = {_aH}_{(t_s - t_1)} \left[\frac{150}{t_s - t_1}\right]^n \tag{3}$$

where $_cH_{150}$ = the calculated, or corrected heat output at the standard temperature difference with steam at 215 F and inlet air at 65 F, Btu per hour
$_aH_{(t_s - t_1)}$ = the actual heat output at the observed temperature difference $(t_s - t_1)$, Btu per hour
t_s = the observed steam temperature, degrees Fahrenheit
t_1 = the observed inlet air temperature, degrees Fahrenheit
n = a numerical exponent

The tests in the warm wall booth were therefore continued for the purpose of consolidating all of the results, and determining whether the heat outputs of all of the types of convectors tested could be represented by an equation of the form of equation (2); and whether some exponent, n, differing from 1.3 used in the correction formula (3) would be more representative of all of the types of convectors tested and produce less individual deviations between the observed and corrected values of the heat outputs at the standard temperature difference of 150 F than the value 1.3 recommended in the Standard Code for Rating and Testing Concealed Gravity Type Radiation (Steam Code). Further tests were also run for the purpose of making a more extensive study of the effect of the height of the cabinet on the heat output and the value of n for two types of convectors.

A previous paper [4] discussed the use of the eupatheoscope for evaluating a complex or non-uniform environment in terms of the equivalent temperature of a uniform environment that could be correlated with human comfort. The principle underlying this instrument involves the measurement of the heat loss

[4] The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933, p. 303.

from a sizable body when the surface is maintained at a constant temperature. The calibration is based on measurements of the heat loss from the eupatheoscope at different air temperatures when both the surrounding walls and the air are maintained at the same temperature. The later use of the instrument implies that for a given uniform environment with walls and air at the same temperature no appreciable change in the heat loss occurs with variations in the relative humidity of the surrounding air. A study was therefore

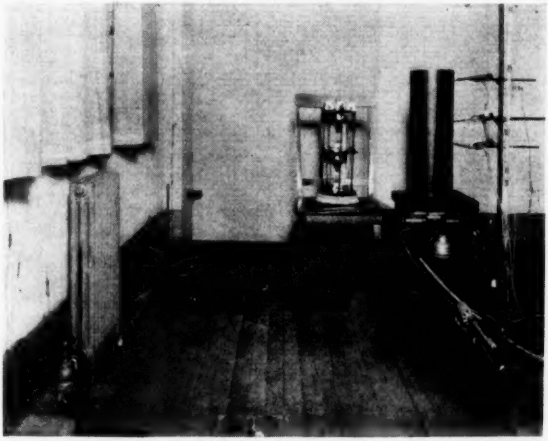


FIG. 1. THE EUPATHEOSCOPE WITH HEATING ELEMENT REMOVED

made in the low temperature testing plant to determine the effect of the relative humidity of the surrounding air on the heat loss from the eupatheoscope. This study was then extended to include the determination of the effect of the relative humidity of the air on the steam condensation or heat output obtained with one type of convector having a non-ferrous heating unit.

DESCRIPTION OF APPARATUS

Both the warm wall testing booth [1, 2, 3] and the low temperature testing plant [1, 4, 5, 6, 7] have been completely described in previous publications.

The eupatheoscope, shown in Fig. 1, consisted of a hollow copper cylinder $7\frac{1}{2}$ in. in diameter and 22 in. long, placed vertically on a stool so that its mean height was 30 in. above the floor. Thirteen thermocouples, made from No. 34 B. & S. gage copper and constantan wire were embedded flush with the surface of the cylinder which was blackened. These 13 thermocouples were electrically but not thermally insulated from the cylinder and were connected in series to

[5] University of Illinois Engineering Experiment Station *Bulletins* Nos. 192 and 223.
[6] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo, A. S. H. V. E. TRANSACTIONS, VOL. 35, 1929, pp. 79-89.
[7] Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock, A. S. H. V. E. TRANSACTIONS, VOL. 38, 1932, p. 351.

a recording-controlling potentiometer, thus affording a means of both indicating and controlling the average temperature of the surface of the cylinder.

In order to heat the surface, current from the line was supplied to a lamp-bank placed inside the cylinder. This lamp-bank, removed from the

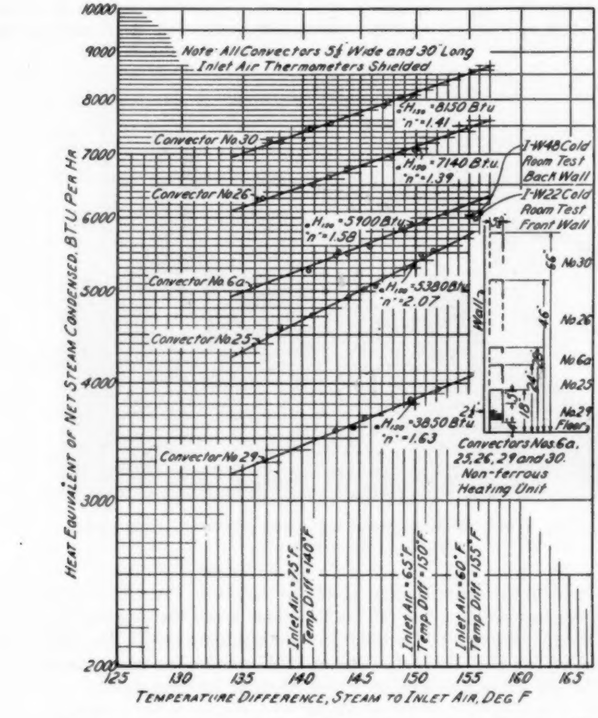


FIG. 2. PERFORMANCE CURVES FOR CONVECTORS NOS. 6A, 25, 26, 29 AND 30 SHOWING EFFECT OF HEIGHT OF CABINET

cylinder, is shown in Fig. 1. The lamps, arranged as shown, in order to give uniform surface temperature were connected in two separate circuits. Parts of the lamps were supplied with a constant amount of current just insufficient to maintain the surface at the required temperature. Current supplied to the rest of the lamps served as an accurate control to maintain the surface at the required temperature, and the total current was measured by means of a sensitive wattmeter.

In all, 7 different types of convectors and an 8-sect., 26-in., 5-tube cast-iron radiator were tested in the warm wall booth. Four of the different types of convectors had non-ferrous heating units, two had cast-iron heating units, and one had a bimetallic heating unit. Two of the non-ferrous heating units were

tested with 5 different heights of cabinets and one cast-iron unit was tested with 2 heights of cabinets. One type of non-ferrous convector was tested with two lengths of heating units, and two types of non-ferrous convectors and one type of cast-iron convector were tested with two different widths of

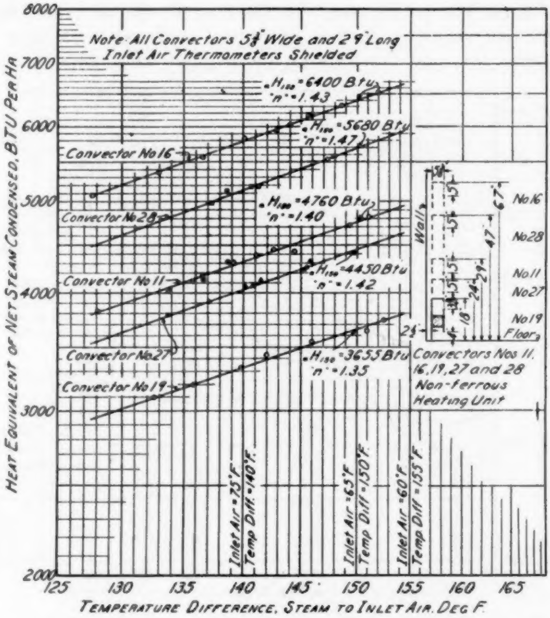


FIG. 3. PERFORMANCE CURVES FOR CONVECTORS Nos. 11, 16, 19, 27 AND 28 SHOWING EFFECT OF HEIGHT OF CABINET

heating units. The dimensions of the different combinations are shown in the diagrams inserted in Figs. 2 to 8 inclusive and in Table 1.

TEST PROCEDURE

All tests were run with a steam temperature of 216.5 F in the convectors. In the case of the warm wall booth, curves were obtained, establishing the relation between the heat output, as measured by steam condensation, and the temperature of the air at inlet, by conducting tests on each convector at different inlet air temperatures varying over a range of from 60 F to 90 F. In order to accomplish this, the large room in which the test booth was erected was heated or cooled to a temperature approximating the desired inlet air temperature, and the convector was allowed to establish the temperature conditions in the booth necessary for thermal equilibrium.

For some of the earlier tests in the warm wall booth the thermometers for

obtaining the temperature of the air at inlet were not shielded against radiation. For later tests these thermometers were shielded in order to obtain the true air temperature. In the cases of the usual types of convectors, equipped with separate cabinets, the correction for radiation to the inlet thermometers did not exceed approximately one per cent, and hence any comparison made from the results would not be affected. In the case of special types, in which the whole front was radiating surface in contact with steam, a large correction would be necessary, and the tests on these types were all run with shielded

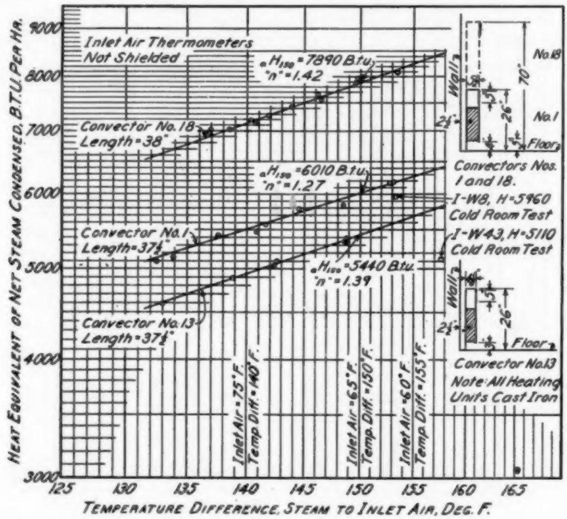


FIG. 4. PERFORMANCE CURVES FOR CONVECTORS NOS. 1, 13 AND 18 SHOWING EFFECT OF HEIGHT OF CABINET

thermometers. In all cases the curve sheets indicate whether or not the thermometers were shielded.

In the few cases in which the convector was also tested in the low temperature testing plant, the temperature in the cold room was maintained at about −2.0 F, and one of the exposed walls was subjected to an equivalent wind velocity of approximately 10 mph. The temperature above the ceiling was maintained at 62 F and the air in the space below the floor at such a temperature that the upper surface of the floor was approximately 2 F warmer than the lower surface. Each convector was allowed to establish whatever temperature conditions were necessary in the room in order to maintain equilibrium between the heat loss from the room and the heat output of the particular convector. In selecting the sizes of the convectors, however, the selection was limited to sizes that would not either overheat or underheat the room an unreasonable amount; that is, temperatures above 75 F or below 65 F at the 30-in. level were not accepted.

For the tests run to determine the effect of relative humidity on the steam

condensation of a convector the temperatures in the cold room and in the spaces above the ceiling and below the floor were varied in order to obtain a range of inlet air temperatures of from 60 F to 70 F when the convector was allowed to establish conditions of temperature equilibrium in the room. A curve of heat output was thus first established with the relative humidities normally obtained without the addition of any water vapor in the room; or

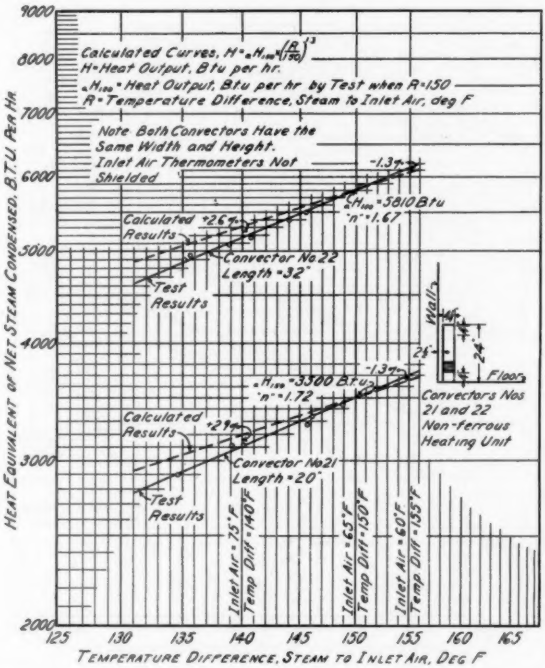


FIG. 5. PERFORMANCE CURVES FOR CONVECTORS NOS. 21 AND 22 SHOWING EFFECT OF LENGTH

with relative humidities ranging from 6 to 15 per cent. Water vapor was then added to the air in the room by evaporating water in an open tank by means of a bayonet type of electrical heater. As the humidity was increased, the convector was allowed to establish temperature equilibrium and the steam condensation was obtained for a number of relative humidities, including a maximum of 68 per cent.

In both the test room and the test booth, no test observations were made until conditions had remained constant for several hours, as indicated by reading of all thermocouples, or thermometers. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per

cent deviation in the successive 10-min increments of weight. At the end of each test, a separate test was made to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

In determining the effect of relative humidity on the heat loss from the eupatheoscope a curve establishing the relation between heat loss and equivalent

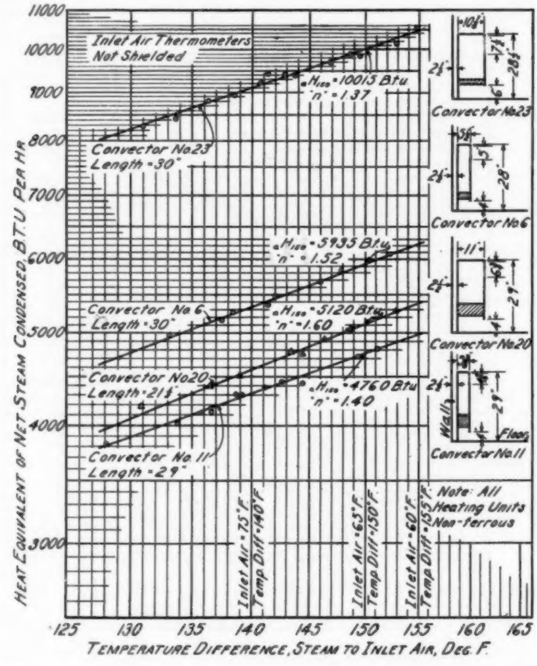


FIG. 6. PERFORMANCE CURVES FOR CONVECTORS NOS. 6 AND 23 AND NOS. 11 AND 20 SHOWING EFFECT OF WIDTH

temperature at a normal relative humidity of 30 per cent, and with the surface temperature of the eupatheoscope at 83 F was first obtained. This was done by placing the instrument in the test room of the low temperature testing plant, and allowing the room to come to temperature equilibrium with the air and the inside surfaces of the walls at the same temperature for a number of different temperatures. No form of heating unit was used in the room for this purpose, but the different equilibrium temperatures were obtained by adjusting the temperature of the air in the laboratory in which the low temperature testing plant was located and then closing the test room and making the observations when equilibrium was established. The relative humidity in the test room was then varied over a range of from 25 to 80 per cent by means of water sprays and an atomizing humidifier in the cold room and the observations were

repeated at several different temperatures, in every case with the wall surfaces and air at the same temperature.

RESULTS OF PERFORMANCE TESTS

The results of all of the performance tests conducted on the convectors in the warm wall booth have been plotted on logarithmic coordinates and are

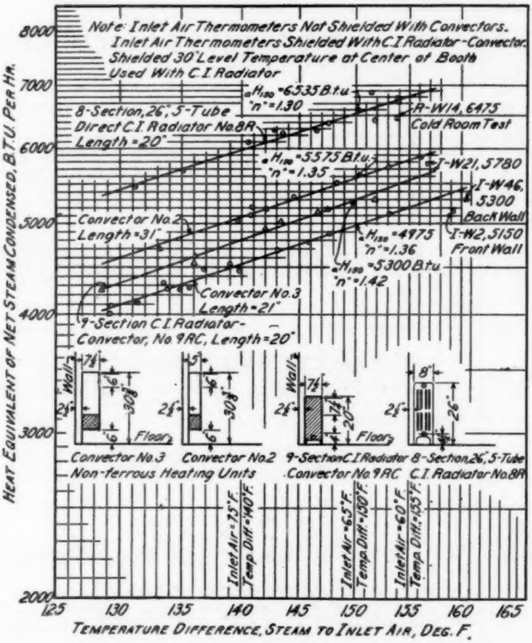


FIG. 7. PERFORMANCE CURVES FOR CONVECTORS NOS. 2 AND 3, C. I. RADIATOR-CONVECTOR, AND DIRECT C. I. RADIATOR

shown in Figs. 2 to 8 inclusive. In Figs 2 to 6 the tests have been classified by groups showing the effects of heights of cabinets and of length and width of the heating units. Figs. 7 and 8 consist of two groups of convectors not subject to any particular classification, but which were included in the general study of the performance of different types. Fig. 7 also includes results obtained with the 8-sec. 26-in. 5-tube direct cast-iron radiator.

Inspection of the curves in Fig. 2 to 8 inclusive indicates that in every case the data can best be represented by a straight line including the whole range of test results. It is therefore evident that for all of the convectors tested, the relation between the heat output and the difference in temperature between the steam and the inlet air can be expressed by a general equation of the form of equation (2) in which n and K assume different numerical values

TABLE 1. DEVIATION OF CALCULATED CORRECTION FROM TEST CURVE

Fig. No.	Con-vec-tor No.	Type of Heating Unit	Overall Dimensions of Convector, Inches			Height of Heating Unit, Inches	Actual (n) from Slope of Test Curve	Heat Output from Test Curve Btu per Hour		Heat Output at 75 F Inlet Corrected to Standard 65 F Inlet			
										By n = 1.3 as per Code		By n = 1.5, Actual Ave.	
			Length	Height	Width			75 F Inlet aH140	Standard 150 F Diff. or 65 F Inlet, aH150	Btu per Hour cH150	Percent Diff. from Test Curve	Btu per Hour cH150	Percent Diff. from Test Curve
1	2	3	4	5	6	7	8	9	10	11	12	13	14
2	29	Non-ferrous	30	18	5½	2	1.63	3440	3850	3765	−2.20	3820	−0.78
2	25	Non-ferrous	30	24	5½	2	2.07	4660	5380	5100	−5.21	5170	−3.90
2	6a	Non-ferrous	30	28	5½	2	1.58	5290	5900	5790	−1.87	5865	−0.59
2	26	Non-ferrous	30	46	5½	2	1.39	6490	7140	7105	−0.49	7200	−0.84
2	30	Non-ferrous	30	66	5½	2	1.41	7390	8150	8090	−0.74	8200	+0.61
3	19	Non-ferrous	29	18	5⅜	5	1.35	3330	3655	3645	−0.27	3690	+0.96
3	27	Non-ferrous	29	24	5⅜	5	1.42	4040	4450	4420	−0.67	4480	+0.67
3, 6	11	Non-ferrous	29	29	5⅜	5	1.40	4320	4760	4730	−0.63	4790	+0.63
3	28	Non-ferrous	29	47	5⅜	5	1.47	5140	5680	5625	−0.97	5700	+0.35
3	16	Non-ferrous	29	67	5⅜	5	1.43	5790	6400	6340	−0.94	6420	+0.31
4	13	Cast Iron	37½	26	4⅛	14³⁄₁₆	1.39	4940	5440	5410	−0.55	5480	+0.73
4	1	Cast Iron	37½	26	5½	14³⁄₁₆	1.27	5500	6010	6020	+0.17	6100	+1.50
4	18	Cast Iron	38	70	5½	14³⁄₁₆	1.42	7140	7890	7820	−0.89	7920	+0.38
5	21	Non-ferrous	20	24	4¼	4	1.72	3105	3500	3400	−2.86	3440	−1.72
5	22	Non-ferrous	32	24	4¼	4	1.67	5170	5810	5660	−2.58	5735	−1.29
6	20	Non-ferrous	21½	29	11	5	1.60	4575	5120	5010	−2.15	5075	−0.88
6	6	Non-ferrous	30	28	5½	2	1.52	5340	5935	5850	−1.43	5920	−0.25
6	23	Non-ferrous	30	28½	10⅞	2	1.37	9110	10015	9980	−0.35	10100	+0.85
7	3	Non-ferrous	21	30⅜	7½	6	1.36	4540	4975	4970	−0.11	5035	+1.21
7	9RC	Cast Iron	20	20	7½	20	1.42	4810	5300	5260	−0.75	5335	+0.66
7	2	Non-ferrous	31	30⅜	5	6	1.35	5080	5575	5560	−0.27	5635	+1.08
7	8R	Cast Iron	20	26	8	26	1.30	5975	6535	6535	0.00	....	....
8	5	Non-ferrous	36	18	5½	2	1.72	4680	5285	5120	−3.18	5190	−1.80
8	7	Bimetallic	22⅝	28⅜	6¹⁄₁₆	20	1.30	5780	6330	6330	0.00	6410	−1.26
8	17	Non-ferrous	26	66	5½	2	1.37	6000	6600	6570	−0.45	6655	+0.83
Average†							1.50				1.24		1.00

† The 8-sect., 26-in., 5-tube, C. I. Radiator, No. 8R, was excluded in obtaining these averages and the percentage differences were averaged without regarding the sign.

depending on the particular type of heating unit or combination of cabinet and heating unit, involved.

Since the performance curves for all of the convectors tested are of the form expressed by equation (2), it is further evident that the formula for correcting the observed heat output under test conditions to an equivalent heat output under standard conditions should assume the form of equation (3); and that

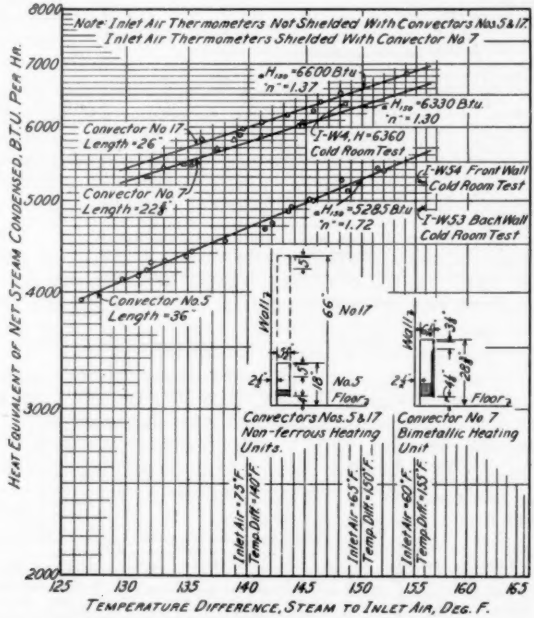


FIG. 8. PERFORMANCE CURVES FOR CONVECTORS NOS. 5, 7, AND 17

the value of n for each particular convector may be determined from the slope of the straight line representing the test data when the latter are plotted on logarithmic coordinates. These values of n have been indicated on the test curves in Figs. 2 to 8. It should be noted in this connection that the temperature difference scale employed in Figs. 2 to 8 is four times the heat output scale, and therefore in order to obtain a graphical solution for the slopes of the different lines, the measured vertical distance should be multiplied by 4.

The present practice in correcting test results as recommended in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[3] is to employ a value of $n = 1.3$ in equation (3) for all types of convectors. For the purpose of comparison, curves having a slope of 1.3 have been shown for convectors Nos. 21 and 22 in Fig. 5. These curves have been made coincident with the actual test curves at points corresponding

to the standard temperature difference of 150 F between steam and inlet air. The deviations between the theoretical curves and the actual test curves at various temperature differences are representative of the errors [1] that would result if the observed heat outputs at these temperature differences were corrected to the equivalent heat outputs at the standard temperature difference

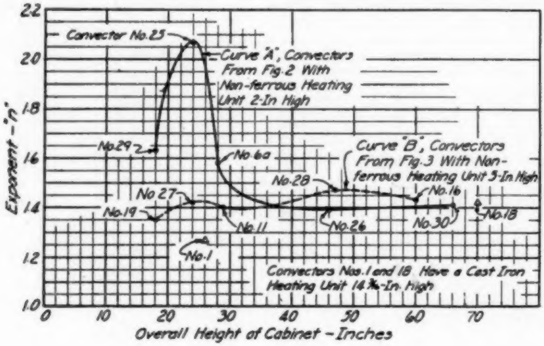


FIG. 9. RELATION BETWEEN EXPONENT n AND HEIGHT OF CABINET FOR THREE TYPES OF CONVECTORS

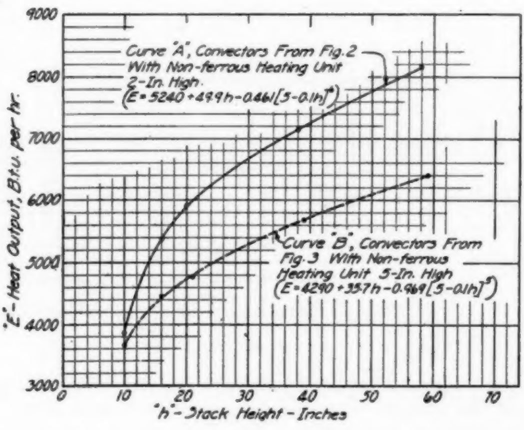


FIG. 10. RELATION BETWEEN HEAT OUTPUT AND STACK HEIGHT FOR TWO TYPES OF CONVECTORS

Note: The equations for curves A and B in Fig. 10 are intended to cover only the practical ratings within the range of those on which tests have been made.

of 150 F. It may be observed that the error becomes greater as the allowable limits for the temperature difference obtained on actual tests deviate further from the standard temperature difference of 150 F. The present practice is to confine the allowable testing range to limits between temperature differences of 140 F and 155 F; or to inlet air temperatures between 75 F and 60 F. It is evident that any method of correction that is acceptable with inlet air at 75 F will also be acceptable for inlet air at 60 F. Hence, this study has been confined to an analysis of the errors resulting when heat outputs observed with inlet air temperature at 75 F are corrected to equivalent heat outputs at the standard inlet air temperature of 65 F with steam at 215 F.

A study of the actual values of n shown by the test data in Figs. 2 to 8, and listed in Table 1, indicates that they vary from a minimum of 1.27 to a maximum of 2.07, but that, excluding the 8-sec., 26-in., 5-tube direct cast iron radiator, 22 values out of a total of 24 were greater than 1.3. Hence, it appears reasonable to assume that a value of n greater than 1.3 should be used in the correction formula applying to all types of convectors if the error is to be reduced to a minimum. The average value for n for all of the convectors tested was approximately 1.5. In Table 1 the equivalent heat outputs at the standard temperature difference of 150 F have been calculated from the actual heat outputs observed at an inlet air temperature of 75 F, as read from the curves at a temperature difference of 140 F, using the values of both 1.3 and 1.5 for n in the correction formula given in equation (3). These calculated values have been compared with the actual heat outputs read from the curves at a temperature difference of 150 F, and the percentage difference between the calculated and the actual heat outputs has been listed in columns 12 and 14. A percentage having a plus sign indicates that the calculated value was greater than the actual.

From column 12 it may be observed that when a correction factor with a value of $n = 1.3$ was used, the equivalent heat output at 150 F temperature difference was, with one exception, less than the actual, and the average error, disregarding the sign, was 1.24 per cent. In the case of column 14 for which a value of $n = 1.5$ was used the equivalent heat output was sometimes greater and sometimes less than the actual, and the average error, disregarding the sign was 1.00 per cent. For $n = 1.3$ there were 7 cases in which the error was greater than 1.5 per cent, and 6 cases in which it was greater than 2 per cent, while for $n = 1.5$ there were only 4 cases in which the error was greater than 1.5 per cent and only one case in which it was greater than 2 per cent. Furthermore, comparing the individual values in columns 12 and 14, it may be observed than in 15 cases out of the 24, excluding the cast-iron direct radiator, the numerical value of the percentage error was reduced when $n = 1.5$ was used, while in 9 cases the numerical value of the percentage error was increased. Hence, from the evidence presented, it seems reasonable to recommend that for convectors a correction factor:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.5}$$

be employed in the A. S. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) instead of the correction factor:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.3}$$

now in use.

The exact significance of the value of n indicated by the performance curves for the different convectors is rather difficult to determine. For free convection, R. H. Heilman[8] shows that the heat loss by convection only is a function of $(t_s - t_1)^{1.266}$. W. J. King[9] states that the heat loss for free convection is a function of $(t_s - t_1)^{1.25}$. The value of n shown for the direct steam radiator,

[8] Surface Heat Transmission, by R. H. Heilman, Trans. *A.S.M.E.*, Fuels and Steam Power Section, Vol. 51, No. 22, September-December, 1921, pp. 287-302.
[9] The Basic Laws and Data of Heat Transmission, III Free Convection, by W. J. King, *Mech. Eng.*, Vol. 54, No. 5, May, 1932, pp. 351-354.

8R in Fig. 7, was 1.30; and the fact that all tests on direct steam radiators consistently show that the heat transmission is a function of $(t_s - t_1)^{1.3}$ probably indicates that the major portion of the heat transfer in this type of heating unit is by free convection and the smaller part occurs by radiation. The combined effect is to increase the value of n from the value of approximately 1.25 for free convection to the value of 1.3 observed for direct steam radiators.

Since radiation constitutes a very small portion of the heat transfer in the case of convectors, it would seem reasonable to expect that the value of n should approach 1.25 if the heat transfer occurred by free convection. The fact that the values of n are practically all greater than 1.35 for convectors may indicate that the stack effect of the cabinet definitely places the heat transfer in the realm of forced convection. In this case the heat transfer will be a function of the velocity of the air through the heating unit, which will depend on relations existing between the frictional resistance through the unit, the stack height, and the temperature of the air in the cabinet; and very little consistency may be expected in the values of n for different types of heating units and different combinations of cabinets and heating units.

In general, types of convectors having performance curves with higher values of n are advantageous in that they represent types in which the heat output increases rapidly as the room cools and the temperature of the inlet air is decreased. A study of the curves in Figs. 2 to 8 and of Table 1 indicates that in general the convectors having thin heating units, or ones in which the vertical distance from the bottom to the top of the heating unit is small, were characterized by performance curves having the higher values of n, while the ones having the thicker heating units had lower values of n.

The effect of varying the height of the cabinet used with the same heating unit, for three types of heating units, is shown in Figs. 2, 3 and 4. As the height of cabinet was increased the total heat output was increased in each case. The values of n for the performance curves also changed. The relation between the value of n and the height of cabinet for the two types of convectors represented by Figs. 2 and 3 is shown in curves *A* and *B* in Fig. 9.

In each case the value of n attained a maximum with some intermediate height of cabinet. This height, for which the value of n becomes a maximum-undoubtedly represents the optimum height of cabinet for the particular heating unit, since at this height the heat output increases the most rapidly per degree Fahrenheit reduction in the temperature of the inlet air. It may be observed from Fig. 9 that with the thinner or lower heating unit (curve *A*) the maximum was attained with a lower cabinet than it was with the higher heating unit (curve *B*). The two points for convectors Nos. 1 and 18 which had the highest heating unit also exhibit this same tendency, although no data on intermediate heights of cabinet were available.

The relative increase in the velocity of the air through the heating units would be determined by the height of the cabinet. The heat transfer would be influenced by the length of the path through the heating unit, or the height of the heating unit. While additional surface added to the height of the heating unit becomes less and less effective, it is probable that as the velocity of the air through the heating unit is increased a critical velocity will be reached above which the rate of heat transfer will decrease more rapidly than the relative rate of increase in velocity unless more heating surface is added. This may

serve to explain why the maximum value of n was attained with lower cabinets in the case of the thin heating units than in the case of the thicker heating units. That is, the critical velocity would probably be lower in the former case than in the latter.

In Fig. 10 the heat outputs at the standard temperature difference of 150 F read from the curves in Figs. 2 and 3 are shown as curves A and B respectively, plotted against the corresponding stack heights. In both cases the observed points fall exactly on well defined smooth curves. This leads to the conclusion that if such a curve can be established with a limited amount of test data for any particular type of heating unit the equation for this curve can be used to obtain the rating of this heating unit with different heights of cabinets without the necessity for obtaining test data on all of the possible heights.

A general type of equation representing both curves shown in Fig. 10 is of the form

$$E = a + bh - c\ (z - h)^m \tag{4}$$

where E = the heat output, Btu per hour
h = the stack height, inches
a, b, c, z, and m = constants determined by the particular type of convector

In the case of the type of convector shown in Fig. 2 and represented by curve A in Fig. 10, equation (4) becomes:

$$E = 5240 + 49.9h - 0.461\ (5 - 0.1h)^8$$

In the case of the type of convector shown in Fig. 3 and represented by curve B in Fig. 10, equation (4) becomes:

$$E = 4290 + 35.7h - 0.969\ (5 - 0.1h)^5$$

The effect of the length of the heating unit with the same height of cabinet is shown in Fig. 5. As the length was increased from 20 in. to 32 in. the value of n for the performance curve was decreased from 1.72 to 1.67. This decrease is small and it is evident from Table 1 that the length had very slight influence on the error resulting from the application of the correction factor.

The effect of the width of the heating unit is shown in Fig. 6. For the type of convector represented by Nos. 11 and 20 an increase in width of the heating unit increased the value of n while for the type represented by Nos. 6 and 23 an increase in the width of the heating unit decreased the value of n. Hence while the width apparently has some influence on the error in the application of the correction factor its effect is small and the nature of the influence is dependent on the type of convector.

In order to make the record of tests complete in this paper certain numbered points representing tests made in the low temperature testing plant or cold room have been shown in Figs. 2, 4, 7 and 8. The heat outputs for these tests may be compared with the heat outputs for the corresponding convectors in the warm wall booth by reading the latter from the curve at the same temperature difference. A complete discussion of these tests has been given in a previous paper and will therefore not be repeated. The conclusion drawn from the tests was that the heat output of a convector under service conditions

in an actual room will be within 10 per cent of the heat output determined from tests in a warm wall booth, with the same temperature for the steam and for the inlet air in both cases, provided that the size of the convector is sufficient to heat the actual room to a temperature of approximately 68 F at the 30-in. level.

Results of Tests at Different Relative Humidities

The results of the tests run to determine the effect of the relative humidity of the surrounding air on the heat output of convector No. 6 are shown in

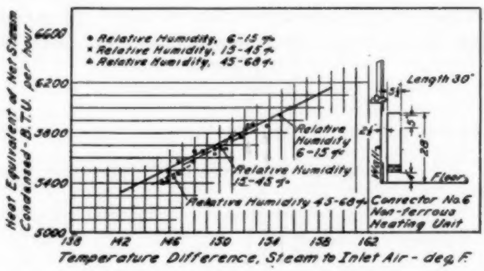


Fig. 11. Effect of Varying Relative Humidities Upon the Heat Loss from a Convector with a Non-Ferrous Heating Unit

Fig. 11. The base curve was determined by the points represented by the open circles and was established for relative humidities varying from 6 to 15 per cent. Three of the points indicated by crosses and representing relative humidities vary from 15 to 45 per cent were on the curve and five points were slightly below the curve. The curve established by these points was approximately 0.5 per cent below the base curve. One of the points indicated by triangles and representing relative humidities varying from 45 to 68 per cent was practically on the base curve and three points were below the curve. A curve established by these points was approximately 0.8 per cent below the base curve. Hence, while the results indicate a slight tendency for high relative humidities to decrease the heat output of a convector with the same temperature difference between the steam and the inlet air, it is evident that relative humidities as high as 68 per cent do not decrease the output to exceed one per cent.

With the standard frame construction used in the walls of the test room it was found impossible to maintain relative humidities higher than 68 per cent with temperatures in the cold room ranging from +2.0 to −6.0 F. Condensation started on the windows when the relative humidity rose to approximately 20 per cent, and began to appear on the inside surfaces of the exposed walls when the relative humidity reached 50 per cent. When the relative humidity exceeded 60 per cent the inside surface of the exposed walls below the 7-ft level was thoroughly wet, water vapor was condensing on the baseboards, and water was running down the panels of the door. The glass in the windows was coated with approximately one inch of ice. The experience with

this room indicates that it would be ruinous to attempt to maintain relative humidities higher than 40 per cent in buildings of frame construction when the outdoor temperature remains at zero, or below zero, for any considerable length of time.

The results of the tests run to determine the effect of the relative humidity of the surrounding air on the heat loss from the eupatheoscope are shown in Fig. 12. From this it is evident that all of the points representing relative humidities from 25 to 80 per cent are on the same calibration curve for the

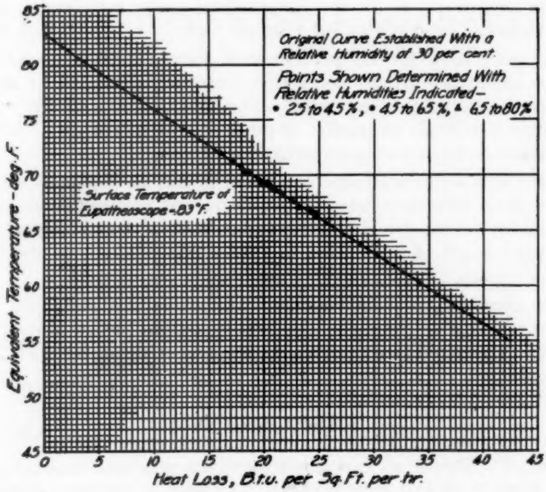


FIG. 12. EFFECT OF VARYING RELATIVE HUMIDITIES UPON THE HEAT LOSS FROM THE EUPATHEOSCOPE

eupatheoscope, indicating that relative humidity has no measurable effect on the heat loss from the instrument. That is, in a given uniform environment with walls and air at the same temperature, no appreciable change in the heat loss from the eupatheoscope occurs with variations in the relative humidity of the air. However, relative humidities of 80 per cent or more affected the insulation of the electrical circuits, and resulted in making doubtful the reliability of the temperatures as observed by means of the thermocouples.

CONCLUSIONS

The following conclusions may be drawn from the test results:

(1) The relation between the heat output and the difference in temperature between the steam and the inlet air for convectors may be expressed by a general equation of the form:

$$H = K\,(t_s - t_1)^n$$

(2) A correction factor $C = \left[\frac{150}{t_s - t_1}\right]^{1.5}$, for reducing the heat output of a convector under test conditions to the heat output under standard conditions, with steam temperature of 215 F and inlet air temperature of 65 F, is applicable within a probable error of 3 per cent for temperatures of air at inlet between 60 F and 75 F.

(3) The error resulting from the use of the exponent 1.5 in the correction factor is less than that resulting from the use of an exponent of 1.3 as now recommended in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[8].

(4) The value of *n* for the performance curves and the magnitude of the error in the application of the correction factor is influenced by the type and physical dimensions of the convector. The exact nature of the error is uncertain, and it is difficult to predict whether it will increase or decrease when the dimensions of the convector increase or decrease.

(5) There is a slight tendency for the heat output of a convector with the same temperature difference between the steam and the inlet to decrease as the relative humidity of the air is increased; but relative humidities as high as 68 per cent do not result in a decrease to exceed one per cent of the heat output obtained with relative humidities of from 6 to 15 per cent.

(6) In a given uniform environment with walls and air at the same temperature, no appreciable change in the heat loss from the eupatheoscope occurs when the relative humidity of the air varies over a range between 25 per cent and 80 per cent.

DISCUSSION

F. B. ROWLEY: This is an excellent presentation of a well rounded out research project. It seems to me that there is sufficient evidence to warrant making the recommended changes in the exponent for the correction formula for tests on convectors. It is also apparent that the question of relative humidity may be neglected for ordinary test purposes.

The relation shown between the output values for convectors of various dimensions should prove valuable in making future designs. The authors of the paper are to be commended for the preparation and presentation of the paper.

No. 999

WHAT IS THE COOLING LOAD FACTOR IN AIR CONDITIONING

By JOHN EVERETTS, JR.† (*MEMBER*), NEW YORK, N. Y.

AIR conditioning, especially summer cooling, is becoming more a necessity than a novelty, particularly in department stores, office buildings, restaurants and other places of business where cooling for comfort is a primary objective.

One of the most important considerations now being given to systems of this type is the operating cost. The forecasting of this operating cost with any degree of accuracy is a complex problem, because it is dependent largely upon the cooling load factor. This factor determines the operating hours of the refrigerating plant required in such an air conditioning system and as this is the most expensive part of the system to operate it is important that this factor be properly calculated. The elements involved in making up the cooling load factor are:

(1) Heat transmission through building
(2) Sun effect
(3) Internal light and power load
(4) People load
(5) Outside air load

The variables that affect the cooling load factors are geographical location and type of installation. The geographical location determines the outside design condition and the portion of the cooling load factor affected by the outside dry bulb and wet bulb temperatures. These outside design conditions for different cities are given in Table 1, Chapter 8, p. 117, The A. S. H. V. E. Guide, 1934. Knowing these conditions and the maintained conditions required, the heat transmission, sun effect and outside air load may be calculated in the recommended manner. It must be borne in mind, however, that the load calculations made under these conditions govern the size of the equipment required to maintain the guaranteed temperature and humidity within the conditioned space at all times. These design temperatures shown in THE GUIDE are not maximum temperatures, nor average temperatures, but they have been chosen after considerable study of the Weather Bureau records as the tempera-

† Engineer, W. L. Fleisher, Consulting Engineer.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buck Hill Falls, Pa., June, 1934.

TABLE 1. SUMMATION FOR YEARS 1925–1931 INCLUSIVE OF HOURLY DRY BULB AND WET BULB READINGS JUNE–SEPT. INCLUSIVE. CENTRAL PARK, NEW YORK CITY

(Data based on 24-hour readings)

Total Hours at Wet Bulb Temperatures Given

*Wet Bulb Temp. F	Average	*Wet Bulb Temp. F	Average	*Wet Bulb Temp. F	Average
81	2	75	42	60–64	710
80	4	74	62	55–59	458
79	7	73	70	50–54	221
78	16	72	121	45–49	72
77	22	71	119	40–44	22
76	37	70	165		
		65–69	778		2,928

Total Hours at Dry Bulb Temperatures Given

Dry Bulb Temp F.	Average	Dry Bulb Temp. F	Average	Dry Bulb Temp. F	Average
101	1	82	72	62	58
100	1	81	72	61	45
99	1	80	95	60	40
98	2	79	81	59	35
97	4	78	106	58	29
96	5	77	104	57	23
95	4	76	125	56	18
94	6	75	134	55	14
93	8	74	147	54	9
92	13	73	159	53	5
91	15	72	160	52	4
90	16	71	144	51	2
89	18	70	158	50	2
88	29	69	125	49	1
87	34	68	133	48	1
86	49	67	106	47	1
85	49	66	104	46	1
84	61	65	87	45	..
83	62	64	79		
		63	71		2,928

* Wet bulb data furnished through courtesy of J. C. Albright, New York, N. Y.

ture that is not exceeded more than 5 to 8 per cent of the time during June, July, August and September for an average year. Therefore, the refrigeration load made up by the transmission, sun effect and outside air is a maximum only 5 to 8 per cent of the season and ranges from zero to maximum 92 to 95 per cent of the time. As these are such variable quantities, it is necessary to determine the average load for the season in order to establish a basis for calculating the operating costs.

Table 1 shows the number of hours during June, July, August and September that different dry bulb and wet bulb temperatures occur. This table has been worked out from the Weather Bureau records of the Central Park Station, in

New York, and is an average of hourly readings taken over a period of the last seven years. Table 1 gives the data for 24-hour operation. The type of system, which may be classified as either industrial process conditioning or comfort cooling, determines which set of tables to use; that is, an industrial process conditioning system operating 24 hours a day to maintain given conditions, would have its cooling load factor based on the data given in Table 1. A comfort cooling installation, such as an office building or a department store,

TABLE 2. VARIATION OF LOAD FROM HEAT TRANSMISSION (INDUSTRIAL PLANT)

Dry Bulb Temp.	Hours	Ton Hours per Season	Dry Bulb Temp.	Hours	Ton Hours per Season
100	1	26.7	86	49	734.0
99	1	25.8	85	49	694.0
98	2	50.0	84	61	812.0
97	4	96.7	83	62	774.0
96	5	116.5	82	72	840.0
95	4	90.0	81	72	780.0
94	6	130.0	80	95	950.0
93	8	166.7	79	81	742.0
92	13	261.5	78	106	883.0
91	15	287.5	77	104	780.0
90	16	293.0	76	125	833.0
89	18	314.5	75	134	780.0
88	29	483.0	74	147	833.0
87	34	538.0	73	159	782.0
			72	160	534.0
			71	144	359.0
			70	158	263.5
			69	125	104.2
				2,060 hours	15,167.1

would use a similar table showing data from 8 a. m. to 5 p. m., whereas a theater cooling system would use a table showing the data from 11 a. m. to 11 p. m.

To show the practical application of these tables in determining the cooling load factor, two examples are given: first, an industrial plant operating 24 hours a day, and the other a department store operating from 8 a. m. to 5 p. m. The design data for the industrial plant in New York City are tabulated as follows:

(a) Outside design conditions	95 Dry Bulb		75 Wet Bulb
(b) Inside maintained conditions	68 Dry Bulb		57 Wet Bulb
(c) Heat Transmission (no sun effect)	270,000	Btu/hour	45.5 per cent
(d) Electric power and light	70,600	"	11.8 per cent
(e) People	45,900	"	7.7 per cent
(f) Outside air	208,000	"	35.0 per cent
(g) Total Load	594,500	"	100 per cent
(h) Tons refrigeration required—49.6 tons.			

To determine the cooling load factor, it is necessary to calculate the total number of ton hours per season that this plant would require. Table 2 shows

the tabulation of the temperature, hours, and ton hours per season for the heat transmission through the building. These calculations have been made for each temperature from 101 F down to the inside maintained temperature at which condition the heat transmission would be zero. The sum total of the ton hours per season for the heat transmission in this case is 15,167.1, as shown.

The heat loads from the lights, motors and people remain constant throughout the season but as they are only effective a total of nine hours daily, the total ton hours per season would be:

TABLE 3. VARIATION OF LOAD FROM OUTSIDE AIR (INDUSTRIAL PLANT)

Wet Bulb Temp.	Hours	Ton Hours per Season	Wet Bulb Temp.	Hours	Ton Hours per Season
81	2	48.6	73	70	1,057.0
80	4	94.0	72	121	1,700.0
79	7	155.2	71	119	1,538.0
78	16	346.0	70	165	1,958.0
77	22	435.0	65–69	778	6,900.0
76	37	685.0	60–64	710	3,083.0
75	42	727.0	57–59	229	185.0
74	62	1,004.0			
				2,384	19,915.8

$$\text{Electric } \frac{70{,}600}{12{,}000} \times 2{,}928 \text{ hours/season} \times \frac{9}{24} = 6{,}460 \text{ ton hours/season}$$

$$\text{People } \frac{45{,}900}{12{,}000} \times 2{,}928 \text{ hours/season} \times \frac{9}{24} = 4{,}210 \text{ ton hours/season}$$

The outside air load is calculated in the same manner as the heat transmission and is tabulated as shown in Table 3, the difference, however, being that the ouside air load from which the total heat must be removed is based on the wet bulb temperature and not the dry bulb. The sum total ton hours per season in this case is 19,915.8, as shown.

The total ton hours per season and their percentages of the total are as follows:

(a) Transmission	15,167.1 ton hours/season	33.1 per cent
(b) Electric	6,460.0 ton hours/season	14.1 per cent
(c) People	4,210.0 ton hours/season	9.2 per cent
(d) Outside air	19,915.8 ton hours/season	43.6 per cent
(e) Total	45,752.9	100.0 per cent

For a 24-hour operation, there are a total of 2,928 hours per cooling season; therefore, the average load would be $\frac{45{,}752.9}{2928} = 15.6$ tons. The refrigeration

required to operate the plant at the design conditions was shown to be 49.6 tons; therefore, the load factor would be $\frac{15.6}{49.6} \times 100 = 31.5$ per cent.

Knowing the operating cost of the refrigerating system per ton of refrigeration per hour at full load; it is merely a matter of simple arithmetic to multiply this cost by the total ton hours per season to arrive at a seasonal operating cost, or if the full load operating cost per hour is known, this may be multiplied by the 2,928 hours per season and the load factor of 31.5 per cent to give the same results.

The second example showing the determination of the cooling load factor for a department store, based on 8 a. m. to 5 p. m. operation, may be calculated in the same general manner as shown for the industrial plant. The data for this particular installation are given below:

Outside design conditions....................	95 Dry Bulb	75 Wet Bulb
Maintained conditions........................	80 Dry Bulb	67 Wet Bulb

HEAT LOAD IN BTU/HOUR AND PER CENT OF TOTAL

	Basement	Per Cent	1st Floor	Per Cent	Total	Per Cent
Transmission.....	45,000	6	286,000	17½	331,000	13½
Sun Effect.......	15,000	2	19,000	1	34,000	1½
People...........	300,000	38	640,000	38	940,000	38¾
Lights...........	200,000	25½	222,000	13½	442,000	17¼
Outside Air......	225,000	28½	480,000	30	705,000	29
	785,000	100	1,647,000	100	2,432,000	100
Tons refrigeration	65		137		202	

Table 4 shows tabulation of the ton hours per season for the transmission load, giving a sum total for the basement of 593.9 and for the first floor 3,737, making a total of 4,330.9, as shown.

The total ton hours per season for sun effect is calculated on the basis that the sun shines only 70 per cent of the 12 possible sunlight hours per day during the cooling season. As the store operation is only nine hours per day, 75 per cent of this value is taken. Calculated in this manner, the total ton hours per season for sun effect in the basement would be 960 and for the first floor 1,218, making a total of 2,178 ton hours per season.

The electric load in a department store would be continuous for the 1,220 hours per season; therefore, the total ton hours per season for the basement would be 20,350 and for the first floor 21,600, making a total of 42,950.

The people load in a department store is quite variable, however, and an average of 60 per cent of the full load capacity may be given as a fair figure. The ton hours per season for the people in the basement on this basis would be 18,300 and in the first floor 39,100, making a total of 57,400.

Table 5 shows the ton hours per season at the different wet bulb temperatures, the sum total being for the basement 5,780 and for the first floor 12,335,

TABLE 4. VARIATION IN LOAD FROM HEAT TRANSMISSION (DEPARTMENT STORE)

Dry Bulb Temp.	Hours	Basement Ton Hours per Season	1st Floor Ton Hours per Season
101	1	5.25	33.10
100	1	5.00	31.80
99	1	4.75	30.20
98	2	9.00	57.00
97	4	17.00	108.00
96	5	20.00	127.00
95	4	15.00	95.00
94	5	17.50	111.00
93	7	22.70	145.00
92	11	33.00	210.00
91	14	38.40	245.00
90	14	35.00	222.00
89	15	34.70	214.00
88	23	46.00	292.00
87	29	50.70	323.00
86	42	63.00	400.00
85	40	50.00	318.00
84	46	46.00	292.00
83	47	35.20	224.00
82	54	27.00	171.50
81	55	13.70	87.40
80	65	0	0
	486 hours	593.90	3737.00

TABLE 5. VARIATION IN LOAD FROM OUTSIDE AIR (DEPARTMENT STORE)

Wet Bulb Temp.	Hours	Basement Ton Hours per Season	1st Floor Ton Hours per Season
81	2	72	150
80	4	129	276
79	6	177	377
78	14	373	798
77	19	456	973
76	31	662	1412
75	30	563	1200
74	45	726	1558
73	42	577	1232
72	67	758	1618
71	49	440	935
70	74	490	1050
69–67	165	357	756
	548	5780	12335
			5780
			18115

making a total of 18,115. The tabulated ton hours per season and their percentages of the total are as shown below:

	Basement	Per Cent	1st Floor	Per Cent	Total	Per Cent
Transmission	594	1.30	3,737	4.75	4,331	3.5
Sun Effect	960	2.00	1,218	1.25	2,178	1.5
People	18,300	40.00	39,100	49.50	57,400	46.0
Lights	20,350	44.20	22,600	28.75	42,950	34.5
Outside Air	5,780	12.50	12,335	15.75	18,115	14.5
	45,984	100.00	78,990	100.00	124,974	100.0

$$\text{Tons refrigeration: } \frac{45{,}984}{1{,}220}=37.7 \quad \frac{78{,}990}{1{,}220}=64.7 \quad \frac{124{,}974}{1{,}220}=102.4 \text{ tons}$$

The refrigeration required as previously shown, to maintain the proper conditions within the store, was 202 tons; therefore, the cooling load factor would be

$$\frac{102.4}{202} \times 100 = 50.5 \text{ per cent.}$$

The operating cost may be calculated in the same manner as previously mentioned.

SUMMARY—CONCLUSIONS

1. As the cooling load factor is made up of many variables, it is impractical to use any one load factor as a criterion for other similar installations unless the items making up the load are approximately the same in proportion.

2. The load factor will be different for similar installations located in different parts of the country because of the difference in the design conditions.

3. The cooling load factor is only to be used to determine the operating cost of the refrigerating end of the system and does not in any way affect the operating cost of other equipment, such as pumps and fans which are generally operated at full load throughout the season.

In Memoriam

Names	Joined the Society	Died
F. Paul Anderson	1921	Apr. 1934
John H. Daly	1915	Mar. 1934
Joseph J. Green	1933	Sept. 1934
William N. McKenna	1912	Jan. 1934
John T. Rothrock	1920	Oct. 1934
Victor N. Welamb	1918	May 1934
Harry A. Wilson	Charter Member	May 1934
Ralph W. Yardley	1920	July 1934
Alex Zech	1904	Aug. 1934

INDEX

A

B

C

D

I

J

K

L

M

N

O

S

W

Y